Arnold Schwarzenegger
Governor

Hot Water Distribution System Research – Phase I

PIER FINAL PROJECT REPORT

Prepared For:
California Energy Commission
Public Interest Energy Research Program

Prepared By:
Applied Energy Technology

November 2005
CEC-500-2005-161

Prepared By:
Applied Energy Technology
Dr. Carl Hiller, P.E., President
Davis, California
Contract No. 500-99-013
Work Authorization No. WA-67

Prepared For:

California Energy Commission
Public Interest Energy Research (PIER) Program

Gary Klein
Contract Manager

Ann Peterson
Program Area Team Lead
Energy Efficiency Research Office

Martha Krebs, Ph. D.
Deputy Director
ENERGY RESEARCH AND DEVELOPMENT DIVISION

B.B. Blevins
Executive Director

DISCLAIMER OF WARRANTIES AND LIMITATION OF LIABILITIES

ORGANIZATION(S) THAT PREPARED THIS DOCUMENT

Applied Energy Technology Company

ORDERING INFORMATION

Requests for copies of this report should be directed to the California State Library, Government Publications Section, Library and Courts Building, 914 Capitol Mall, Sacramento, CA 95814

CITATIONS

This report was prepared by

Applied Energy Technology
35 College Park
Davis, CA 95616

Principal Investigator
C. Hiller

This report describes research cosponsored by The California Energy Commission and Applied Energy Technology Company.

The report is a corporate document that should be cited in the literature in the following manner:

Hot Water Distribution System Research – Phase I Final Report, California Energy Commission Report No. XXXXX, May 2005.

REPORT SUMMARY

Background

Hot water distribution (HWD) systems dramatically influence the amount of time, water and energy wasted while delivering hot water from water heaters to fixtures, thereby having important impacts on total water heating system energy use. Surveys of existing literature showed that little research had been done to quantify time, water, and energy waste characteristics of different HWD system piping configurations. Moreover, surveys of existing HWD models showed that none had ever been rigorously validated against actual test data, and that existing models relied either on simple plug-flow theory or fully-developed flow theory, neither of which was probably valid in the developing flow conditions that characterize real hot water draws. Most disturbing of all, interviews with many new residential building owners revealed that hot water delivery times and water waste have been getting steadily worse with newer buildings. For this reason, the California Energy Commission (the Commission) embarked on a multi-faceted effort to improve knowledge about HWD system performance, aimed at updating building energy efficiency standards with respect to HWD systems. The work described here quantifies the time, water, and energy waste characteristics of the most commonly seen HWD piping systems under a variety of conditions.

Objectives

The objectives of this effort were to:

- Perform a brief survey of HWD system design and installation practices in Northern California by conducting on-site inspections and interviews.
- Perform laboratory testing on the most commonly seen HWD system piping materials and sizes in order to quantify pipe heat loss characteristics and time, water, and energy waste during the delivery of hot water to fixtures, as functions of important variables.

Approach

First, a series of residential construction site visits were conducted in order to learn what HWD system designs, installation practices, and installation conditions were occurring in the field. Second, based on the site-visit observations, a set of laboratory tests were designed and performed, quantifying water and energy waste characteristics of the types of HWD piping mostly commonly seen in the field. A test laboratory was established in which to perform these tests on full-size piping systems.

Results

The most important findings of this work are:

- Analysis of site visit data shows that at least for the sites visited, length and volume of under-slab piping was greater than expected, and greater than would be typical of above-ground piping. A method of estimating piping lengths and volumes for "conventionally" laid-out HWD systems is provided in this report. "Conventional" in this case means piping that runs parallel and perpendicular to foundation walls and other structural members, as opposed to diagonally. This estimation method can be useful for comparing alternative HWD system piping layouts.

- It appeared that the under-slab environment is always at least damp, and sometimes wet to the point of having standing water in pipe trenches. Whether or not the under-slab environment stays damp over time is unknown. Tests performed here indicate, at least qualitatively, that moisture presence in the under-slab environment can substantially increase heat loss from uninsulated hot water distribution system pipe in that environment. It is unknown whether or not the under-slab environment will over time become a thermal storage medium that may lessen heat loss from buried pipes. Performance of insulated pipe in the under-slab environment has not yet been investigated.

- Regarding actual pipe insulation practices observed at sites visited, insulation treatment at elbows and Ts could have been much better. Additionally, butt-joints of insulation segments would perform better if glued together as per manufacturer recommendations. Moreover, insulation segments should be axially compressed during installation to compensate for insulation shrinkage over time, as per manufacturer recommendations. These practices were not seen at any of the sites visited.

- Laboratory testing and analysis showed that the extent to which pipe insulation impacts HWD system energy waste depends on the environment where the pipes are run and the nature of the draws, especially regarding time-spacing of draws. Pipe insulation provides dramatic improvements in pipe cool-down rates, extending cool-down times by 200-400% compared to bare pipe. Slowing pipe cool-down can save significant HWD system energy loss by significantly reducing the need to purge piping of luke-warm or cold water before draws. With pipe insulation, the water in the pipe will remain above a useful temperature between many draws in real situations. Adding pipe insulation on pipes in recirculation-loop HWD systems saves significant amounts of energy if run-time of the RL system pump(s) is at all significant (i.e. greater than one hour, possibly less). The pipe heat loss (UA) factors reported here allow estimation of piping heat loss rates, pipe temperature vs time during cool-down, and more.

- Laboratory testing also showed that ½ inch thick foam pipe insulation performs almost as well as ¾ inch thick foam pipe insulation, providing heat loss rates that were within 6% of those of the ¾ inch thick insulation. The thicker insulation provided slightly greater improvements in cool-down time, ranging from 7-13% longer than for ½ inch thick foam.

This difference in cool-down time could be important in real installations, depending on draw time-spacing.

- Time, water, and energy waste during delivery-phase flow have been quantified as functions of important variables in this report. The important variables include hot water temperature, initial pipe temperature, ambient temperature, pipe size, pipe length, flow rate, insulation level, and pipe material.

- It was discovered that flow during the delivery phase of a hot water draw is a highly transient, non-steady-stage phenomenon, where hot and cold water are flowing simultaneously in the same pipe. Flow during the delivery phase (and in some cases continuing for prolonged periods of draw thereafter) was discovered to be characterized by at least 3 distinctly different flow regimes. We have termed these flow regimes stratified flow, normal flow, and slip-flow.

- At low flow rates and low initial pipe temperatures, where flow of the cold water in the HWD pipe during the draw was laminar, flow stratification was observed to occur, where hot water would flow a greater distance down the top side of the pipe than on the bottom. Flow stratification increased water and energy waste by allowing more heat transfer to occur between hot and cold water flowing in the pipe than if the stratification had not occurred. This was especially true in pipe entrance regions. Axial heat conduction in the pipe wall also contributed to this detrimental hot-to-cold water heat transfer at low flow rates.

- At intermediate flow rates, hot water temperatures, and initial pipe temperatures, water and energy waste during the delivery phase were primarily influenced by mixing of hot and cold water in the pipe caused by flow turbulence, and by heat transfer from the hot water to heat up the mass of the pipe wall. This was the “normally occurring” mode of delivery-phase flow.

- At high flow rates and warm initial pipe temperatures, slip-flow tended to occur in the pipes. Slip-flow was characterized by shearing of the boundary layer, reducing the turbulent eddies that caused mixing in the pipe. Slip-flow reduced water and energy waste by reducing both heat transfer to ambient, and by reducing mixing of hot and cold water in the pipe. Slip flow was observed to occur more readily at lower flow rates and temperatures in some types of piping compared to others. More study of piping system configurations and materials is needed to better understand the slip-flow phenomenon and how to capitalize on it.

- Heat loss to ambient was found to significantly affect water and energy waste under low flow rates for fairly long lengths of uninsulated pipe. However, these pipe lengths and flow rates are in the range of what is commonly seen in real HWD system installations so this is a significant effect. Moreover, this effect becomes highly significant even at higher flow rates and shorter pipe lengths if the piping is in a high heat-loss environment, such as in very cold surroundings, moving air, or wet conditions.

- Time, water, and energy losses associated with vertical piping appeared similar to those for horizontal within the narrow range of vertical pipe configurations tested. The most notable difference was lack of a stratified flow regime. In vertical pipe flow stratification is replaced by full-diameter mixing, similar to "normal" flow mixing. Some unexpected behavior was observed regarding vertical natural convection flow in vertical pipes. Stable inverted temperature gradients were observed to occur in vertical pipe exposed to hot water at the bottom. It appeared that heat transfer in the pipe material could cool the warm water enough to stall vertical natural convection flow. More investigation of this phenomenon is recommended.

- Examples are provided showing how to use the pipe heat loss and delivery-phase time and water waste findings to calculate HWD system energy losses. Tank standby heat loss is important in these energy waste calculations because some HWD system piping configurations require higher tank setpoint temperatures than others in order to deliver usably hot water at fixtures. Since tank heat loss occurs 24 hours per day 365 days per year, even small increases in tank setpoint temperature can have important energy impacts.

Significance

This research makes available quantitative HWD system performance data that can be used to evaluate and upgrade HWD system performance models. Additionally, this work can be used to analyze HWD system performance as an aid to updating California's Residential Building Energy Efficiency Standards with regard to HWD systems. Equally important, this work can be used to develop improved HWD system design manuals and processes.

Keywords

- Water Heating
- Hot Water Distribution
- Heating
- Piping
- Energy
- Water

ABSTRACT

Prior to this effort, little information was available on the actual performance of hot water distribution (HWD) systems. Moreover, the few HWD system models in existence relied on assumptions that were not valid under the developing flow conditions that characterize flow in HWD system piping, and those models had never been rigorously verified against real piping system data. Most disturbing was the fact that discussions with many new residential building owners revealed a general trend toward worsening time, water, and energy waste characteristics of HWD systems in newer buildings. For this reason, the California Energy Commission embarked on a multi-faceted effort to improve knowledge about HWD system performance, aimed at updating building energy efficiency standards with respect to HWD systems. The work described here quantifies the time, water, and energy waste characteristics of the most commonly seen HWD piping systems under a variety of conditions.

A series of site visits were conducted to identify common HWD system piping practices and installation conditions. Based on the site observations, a dedicated HWD system test laboratory was established, where tests were performed on the most commonly seen HWD piping configurations. The time, water, and energy waste characteristics of those systems were quantified, and the results are presented in this report. The information in this report is useful for improving the design and analysis of HWD systems, and will be extremely useful in the setting of improved building energy efficiency standards with respect to HWD systems.

ACKNOWLEDGEMENTS

The principal investigator would like to acknowledge Gary Klein of the California Energy Commission for his encouragement, guidance, and support of this project, and the members of the Commission's Hot Water Distribution Core Team for their review and commentary on the work performed.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1 INTRODUCTION

Background

Hot water distribution (HWD) systems dramatically influence the amount of water and energy wasted while delivering hot water from water heaters to fixtures, thus having important impacts on water heating system energy use. Moreover, they also dramatically affect user satisfaction by influencing the amount of time spent waiting for "hot-enough" water to arrive at fixtures. Surveys of existing literature showed that little research had been done to quantify time, water, and energy waste characteristics of different HWD system piping configurations. Moreover, surveys of existing HWD models showed that none had ever been rigorously validated against actual test data, and that existing models relied either on simple plug-flow theory or fully-developed flow theory, neither of which was probably valid in the developing flow conditions that characterize real hot water draws. These shortcomings prompted the California Energy Commission (the Commission), with cosponsorship by Applied Energy Technology Company (AET), to initiate a series of research projects aimed at developing a comprehensive plan to address HWD issues. The ultimate goals are to produce information and tools to facilitate informed HWD system design and energy analysis and to guide codes and standards setting processes. The effort reported on here specifically addresses determination of how HWD systems are currently being designed and installed in California, and additionally focuses on producing quantitative performance data on of a variety of HWD system piping configurations under controlled laboratory environments.

Goals and Objectives

The objectives of this effort were to:

- Perform a brief survey of HWD system design and installation practices in Northern California by conducting on-site inspections and interviews.
- Perform laboratory testing on the most commonly seen HWD system piping materials and sizes in order to quantify pipe heat loss characteristics and time, water, and energy waste during the delivery of hot water to fixtures, as functions of important variables.

2
METHODOLOGY

Site Visits

Both the Commission and AET project managers identified possible locations for site visits to inspect installation of various types of HWD systems. The site visits covered the following types of structures and HWD systems. All sites visited used slab-on grade construction, and most, but not all, used HWD piping under the slab:

- Tract-built single family detached homes
- Tract-built single family attached (duplex, townhouse) homes
- Custom-built single family detached homes
- Medium-to-high density (apartment/condominium) multi-family homes
- Copper, plastic, and plastic/aluminum/plastic piping
- Conventional trunk-and-branch
- Multiple parallel branch-mains
- Manifold system with dedicated pipes to each fixture

Laboratory Testing

A laboratory test plan was developed based on the types of HWD piping systems and environments seen in the site visits. The AET water heating test laboratory was moved and modified to accommodate testing of various full-scale HWD system piping configurations. Tests were then performed, beginning with the most common types of piping systems. The test plan was modified as testing proceeded and results were obtained. A variety of improvements to the laboratory test apparatus were implemented during the testing that allowed a larger number of tests to be performed more quickly than originally envisioned, resulting in a greater number of tests than required by the contract.

3
SITE VISIT RESULTS

Table 3-1 shows a matrix of the numbers of the various types of HWD system building types and piping systems that were observed during the site visits. The contract called for visiting at least two construction sites in at least two counties. A total of 20 sites were actually visited, in three different counties. The counties were Sacramento, Yolo, and Contra-Costa, California. Pictures were not taken at some of the sites because their piping practices appeared the same as previous sites. A CD containing the site visit pictures was submitted as one of the contract deliverables and is available separately from the Commission.

Table 3-1
Site Visit Piping Systems vs Building Type

PIPING TYPE	SINGLE FAMILY DETACHED	SINGLE FAMILY ATTACHED	MULTI-FAMILY
Rolled Copper	4	4	
Rigid Copper	4	4	
PEX-AL-PEX	6	4	1
PEX/MANIFOLD	1		

General Observations

All of the sites visited were slab-on-grade construction. Most buildings employed hot and cold water piping under the slab. However, at least one major tract-builder did not use any under-slab water piping. Moreover, in multi-story structures, a higher percentage of the piping is above the slab than in single-story under-slab piping systems.

Plumbing codes require that there be no fittings in under-slab piping. This means that rigid pipe, which would need couplings and elbows, cannot be used under slabs. Only continuous rolled pipe is used under slabs, with all connections made in the walls above the slabs.

Tract builders that employed under-slab piping for the most part used PEX-Aluminum-PEX (commonly referred to as PAX or XPA piping), both under and above the slab. This piping, as shown in figure 3-1, uses an inner layer of high-density cross-linked polyethylene (PEX), a thin middle layer of aluminum (as an oxygen barrier), and an outer layer of PEX. The hot water PAX piping was normally color-coded orange, and the cold water PAX piping was normally blue. The tract builder that did not use under-slab piping still used PAX piping above ground. PAX piping uses drilled-brass connection couplings with special banded compression fittings, as seen in figure 3-2. Interestingly, wherever PAX piping penetrated through walls such that the pipe or connecting fittings would be visible, it was connected to copper or brass fittings for the visible part, as shown in figure 3-3. This means that when the house is completed, the piping system looks as if it were a copper piping system instead of PAX.

It should be noted that PAX piping has not been in use in significant quantities long enough in HWD system applications to draw conclusions regarding its longevity compared to copper. It will need to have been in use for 40-60 years before meaningful comparisons can be drawn regarding longevity.

The custom-built homes visited for the most part used rolled-copper piping under the slab and rigid copper piping above the slab.

Pipe Insulation Observations

Piping sizes seen at the sites were mostly nominal ¾ inch and ½ inch. Some 1 inch was seen, particularly in the multi-family buildings, and on main PAX lines from water heaters to where the multiple parallel main branch lines began, as seen in figure 3-2.

Where piping passed through cement in under-slab piping installations (footings, foundation walls, slab), or where it would be directly covered by soil or gravel without a sand protective layer, the piping was either insulated, or was sheathed in thin color-coded polyethylene plastic, approximately 6 mil (.006 inches) thick. As shown in figures 3-2 and 3-4, blue was generally used for cold water and orange or red for hot water. The main function of the thin plastic covering is to allow the pipe to slide within the surrounding soil, gravel or cement to accommodate thermal expansion and contraction, thus reducing wear on the pipe. Rubbing wear caused by pipe thermal expansion and contraction is one of the failure modes of under-slab piping systems. In general, under-slab piping should not be buried directly in gravel because of the potential for wear, even when using the thin protective plastic sheaths. However, it appeared that at some sites some parts of the pipes were in direct contact with gravel – a poor installation practice.

The site visits were conducted in the January –March 2004 time frame. It appeared that in tract-built single- and multi-family homes where the slabs were poured prior to January 1, 2004, none of the PAX under the slab was insulated, as shown in figure 3-2. In similar slabs poured after January 1, 2004, only the piping from the water heater to the main kitchen sink was insulated, and only if the nominal pipe size was ¾ inch or over (and most were). All other piping was left uninsulated, both below and above the slab, as shown in figures 3-2 through 3-8.

In the custom-built homes, almost all of the hot water piping, both above and below the slab, was insulated, as shown in figures 3-9 through 3-12. Note that there were occasionally sections of under-slab hot water pipe that were still not insulated, as seen in figures 3-9 and 3-10.

Pipe insulation seen in the field was exclusively black closed-cell foam, either polyolefin or polyethylene, with thicknesses of either ½ inch or ¾ inch. As shown in figure 3-13, most often the dimensions and R-value were marked on the outside of the insulation. Table 3-2 shows R-value ratings (hr ft^2 F/Btu) vs insulation thickness as observed on pipes at the sites. Manufacturers' literature typically claimed that the thermal conductivity of the foam insulation was approximately k=0.02 Btu/hr ft F. One can directly calculate the effective R-value of pipe insulation made of these materials by using the above thermal conductivity value, the inner and outer insulation diameters, and using the outer diameter as the basis for calculating the R-value. Most introductory heat transfer textbooks explain radial heat conduction calculations associated with heat transfer from pipes (see references).

Manufacturers' literature indicated that the observed pipe insulations are approved for direct burial above the water table, with a recommendation that they be backfilled with sand to prevent damage.

Table 3-2
Pipe Insulation R-Values (hr ft^2 F/Btu)

NOMINAL PIPING SIZE (in.)	½ inch FOAM (k=.02Btu/hr ft F)	¾ INCH FOAM (k=.02 Btu/hr ft F)
½	3.1	5.2
¾	2.9	4.7

In the above-slab piping insulation (only seen in custom homes), insulation treatment at elbows and pipe T's was generally not very good, as shown in figures 3-11 and 3-12. Elbows were usually left either totally uninsulated, or open on one end. Bevel cuts (45 degree cuts), which are one simple way to form good insulation closure at elbows (see test fixture pictures in appendices C, and D), were not seen at any of the sites visited.

Moreover, manufacturers recommend that the butt-ends of the insulation pieces be bonded together with an approved adhesive, especially in under-ground applications. This was not seen at any of the residential sites visited for this project, as shown in figure 3-13.

A shrinkage factor of around 4% was usually listed for the pipe insulation observed at the sites visited. This shrinkage is most likely due to outgasing as the insulation is heated, and is irreversible. This means that eventually, as the total time spent under hot conditions increases, each pre-cut 6 foot length of insulation will shrink by almost 3 inches. Because of this the manufacturers recommend that during installation, insulation pieces should be axially compressed at least 2-3 inches per 6 foot length, to allow for eventual shrinkage under hot operating conditions. This axial compression was generally not seen in practice. The lack of axial compression and butt-end bonding means that over time, axial gaps about 2 inches long may develop in the insulation at roughly 6 foot spacings. Exact amount of shrinkage is a function of the pipe temperature and duration at temperature.

Under-Slab Piping Practices

A number of under-slab piping practices that adversely affect water and energy waste in HWD systems were observed. One of the most important generalizations derived from these observations is that under-slab piping is much longer than would appear from merely looking at fixture locations:

- Since no fittings are allowed under slabs, all piping must go above the slab to connect to a fixture and the next section of piping, and then back under the slab again to go to the next fixture. This means that under-slab piping is always continuous rolled piping of some sort, usually rolled copper, PEX or PAX. It also means that piping length is substantially longer than one would expect because of the extra vertical piping at each end of the continuous length of pipe. Also, at each fixture point there is typically a sharp 180 degree fitting that would increase pressure drop and cause turbulence and mixing inside the pipe. See figures 3-2 through 3-10.

- HWD piping is usually run in the same trenches as the drain piping of the fixture being served. This means that HWD piping normally runs parallel to foundation walls, as shown in figures 3-6 and 3-9, rather than diagonally This also means that under-slab piping is always much longer than one would visualize by just measuring the shortest horizontal distance between fixtures.

- Rolled piping must use gradual bends to avoid kinking the pipe. This means that HWD piping must be buried a substantial distance beneath the slab so that the gentle curve can be made prior to the pipe rising vertically through the foundation wall. To reduce thermal expansion and contraction wear, generally only straight piping without bends is seen in cement. Piping depths of a minimum of 2 feet were observed, but depths were usually closer to 4 feet, as seen in figures 3-4 through 3-6 and 3-9. Piping most often rose into load-bearing walls under which there is a substantial vertical depth pf concrete to act as the footing and foundation. Where piping rose into load-bearing walls it typically entered through the footing and foundation such that it traveled vertically through at least 2 feet of concrete. Where piping rose to fixtures such as kitchen island sinks, piping rose through only 3-4 inches of concrete.

- Because of the need for gentle curves in the rolled piping, piping between relatively close fixtures was always much longer than one would think, because the piping had to bend both vertically and horizontally to travel between fixtures, as seen in figures 3-5and 3-9.

- Under slab HWD piping is substantially longer than one would estimate from just looking at fixture locations and shortest horizontal spacing for yet another reason. Since connections are made above the slab, hot water flow to the furthest fixtures usually flows in series through connections at each other fixture along the pipe, rather than having parallel branches off a main trunk line. Instead of having one main trunk line and multiple side branch lines serving each fixture, under-slab HWD piping usually has multiple parallel branch lines, with flow in series through each successive fixture along the branch lines. This can be seen in figures 3-4 through 3-6 and 3-9. Note that this practice sometimes causes piping to double back on itself, such that fixtures that appear very close to each other are actually far apart from a piping length viewpoint, as shown in figure 3-9.

All of the above observations reveal that under-slab piping is substantially longer than one would expect based on an assessment of horizontal fixture distances. While extensive under-slab piping length measurements were not taken, some spot-measurements were. Based on these spot measurements, and examination of pipe lengths from the many pictures taken, table 3-3 gives a guideline for estimating actual pipe length between fixtures in under-slab HWD piping. Note that it is important to know which fixtures are connected to which other fixtures and in what order to correctly use table 3-3. The multipliers in table 3-3 were determined from the site visit data using the empirically derived formula:

$M=(.004444)(X-20)^2 + 1.5$ for $X<20$ feet

$M = 1.5$ for $X \geq 20$ feet

Where X = shortest horizontal distance between fixtures measured in feet

Table 3-3
Actual Under-Slab Pipe Length vs Shortest Horizontal Distance Between Fixtures

SHORTEST HORIZONTAL DISTANCE BETWEEN FIXTURES (FT)	PIPE LENGTH MULTIPLIER	ADDER FOR VERTICAL ENDS RISING ABOVE SLAB (FT)	TOTAL ESTIMATED PIPE LENGTH BETWEEN FIXTURES (FT)
5	2.5	2 ft each end	16.5
6	2.37	2 ft each end	18.2
7	2.25	2 ft each end	19.8
8	2.14	2 ft each end	21.1
9	2.04	2 ft each end	22.4
10	1.94	2 ft each end	23.4
15	1.61	2 ft each end	28.2
20	1.5	2 ft each end	34
25	1.5	2 ft each end	41.5
30	1.5	2 ft each end	49
35	1.5	2 ft each end	56.5
40	1.5	2 ft each end	64
50	1.5	2 ft each end	79
60	1.5	2 ft each end	94
70	1.5	2 ft each end	109
80	1.5	2 ft each end	124

Note: Because under-slab piping tends to run in multiple parallel branch lines which serve fixtures in series, actual piping lengths from the water heater to fixtures are quite long – much longer than just the pipe lengths between fixtures shown in table 3-3. The examples discussed at the end of this section illustrate this point.

Another noteworthy observation is that in all but custom homes, uninsulated hot and cold water pipes often passed in close proximity to each other (usually 2-6 inches) when passing through concrete, as seen in figure 3-2. Since concrete is a fairly good thermal conductor, this could allow significant heat transfer between the hot water pipe and the concrete, and also between the hot and cold water pipes through the concrete. Both increase water and energy waste.

Under-Slab Environment

Piping manufacturers and many local plumbing codes recommend that piping be buried in sand backfill before returning the original soil to the trench. This helps protect the pipe from crimping against hard soil and wear due to thermal expansion and contraction. The site visits revealed that often the backfill is other than ideal. Usually the soil from the trenches is piled near the trench, and chunks of this soil fall back into the trench during construction, covering the piping in a non-uniform manner. Sometimes sand is not used at all and the pipe is simply covered with the original soil from the trench. Construction workers walking in the trenches pack some of the soil tightly around the pipes, while in other areas there are large air spaces. Additionally, as seen in figure 3-8, after rains, gravel is sometimes spread on the soil before the sand and sub-slab materials are placed, to reduce having to walk in mud. This gravel sometimes finds its way into direct contact with the under-slab pipes.

The most important observation was that the under-slab environments seen during the site visits were always at least damp, and were sometimes completely saturated to the point of having standing water in the trenches. At many of the sites visited, rain had occurred one or more times during preparation of the under-slab environment but before the slab was poured. When rain occurred, the ground under where the slab would be was always thoroughly wetted. If the soil had high clay content, water usually did not percolate well through the soil and the trenches typically filled with several inches of water. Figure 3-10 shows puddles of standing water typical of such sites. There is an extreme amount of mud at building sites when it rains before the slab is poured.

Additionally, even if rains had not occurred, the under-slab environment is made to be wet or damp during the construction process. Before concrete is poured, especially before the slab itself is poured, the normal construction practice is to heavily water the earth, sand, and gravel under where the slab will be. Interviews with on-site construction personnel revealed that this is done for two reasons. First and foremost, it is done because it is necessary to ensure proper curing of the concrete. When concrete is poured it does not have 100% of the water content needed for full chemical reaction to cure the concrete. This is because adding 100% of the necessary water would cause the concrete to shrink and crack as it cured. Instead, water is added to the materials below the slab before the pour to provide extra moisture for the concrete curing process. Similarly, the top-side of the slab is often watered several times over several days to provide some of this extra moisture needed for curing. A second reason given for heavily watering the under-slab environment before concrete was poured was to cause the soil to expand. This is apparently especially important in clay-rich soils. Clay soils expand substantially when wetted. By pre-wetting the soil under the slab, the soil expansion is forced to occur before the concrete is poured, hence eliminating the possibility of cracks opening up in the concrete as the soil expands upon later wetting. If the soil contracts after the slab is poured, the slab settles downward, causing the slab to go into compression, rather than tension, and hence keeping closed any cracks that form.

Above-Slab Piping

Above-slab piping, which is run inside walls and floors, and between ceiling joists, is almost always run either parallel or perpendicular to studs and joists. It is rarely run diagonally. This means that above-slab piping is also substantially longer than one would anticipate from just examining the horizontal distances between fixtures.

Single Story With Under-Slab Piping

In single-story structures, if HWD piping was under the slab, there was typically little above-slab piping. The distance from top of the slab to the fixtures was typically around 2-3 feet for sinks and other fixtures, except showers. For showers the distance was more like 4 feet to the mixing valve plus another 2 feet to the shower head from the mixing valve, or for tub/shower combinations 2 feet to the mixing valve and another 4 feet to the shower head.

There is also additional piping length getting the hot water from the water heater into the slab. In California, the majority of water heaters are currently installed in garages, where they are raised at least 18 inches above the floor. Total hot water pipe length from top of the water heater down to the slab is hence 7-8 feet, including some horizontal length to clear the top of the water heater.

Single Story With No Under-Slab Piping

In single-story structures without under-slab piping, HWD piping was typically run from the water heater up to the attic, and then distributed in a main trunk line with multiple parallel branch lines to each fixture. Vertical pipe lengths in walls in this case were more or less the reverse of the case with under-slab plumbing, assuming an 8 foot story height (this varies with architectural characteristics of the structure). Vertical drops to showers were typically at least 4 feet to the mixing valve and then 2 feet back up to the shower head, and to tub or tub/shower combinations were more like 6 feet down to the mixing valve and then 4 feet back up to the shower head. Vertical drops to sinks and most other fixtures were typically on the order of 6 feet. Risers from the water heater to the attic were around 2-3 feet in length.

One notable difference between under-slab and above-slab piping was that for double sinks, such as in the master bath, piping lengths between sinks were much shorter in above-slab piping than in under-slab piping. In under-slab piping, a long loop of piping typically carried water from one sink to the next, with its associated 2 to 4 feet of extra vertical length at each end plus generous extra length to accommodate the gentle under-slab pipe bend. In above-slab hot water piping, the sinks were usually directly connected by a short length of straight pipe traveling horizontally through the wall studs. This meant, for example, that for a 5 foot spacing between sinks, the above-ground hot water piping between the sinks was about 5 feet long, while the below-slab piping between the two sinks was on the order of 16.5 feet (see table 3-3) plus a two foot riser from the slab to the sink plus a one foot riser for the above-slab connection for the two below-slab pipes. The total piping length between the two sinks in the below-slab case was hence about 19.5 feet, compared to 5 feet for the above-ground piping.

While extensive measurements were not taken on most above-ground piping of the trunk-and branch type, some spot measurements were taken. Distances from water heaters to fixtures can be roughly estimated as 1.5 times the shortest horizontal distance, plus the vertical distances noted above. Note, however, that HWD main trunk-line pipe sizes typically drop in size, as the line extends past other fixtures. One must know where the pipe size changes in order to accurately estimate amount of water in the pipe for delivery-delay and water and energy-waste calculations. When estimating actual pipe volumes from water heater to fixtures in above-ground piping, it is best to estimate where the pipes actually run in the structure, and then estimate the length and volume of each portion as the pipe changes size. The examples from appendix A which are discussed at the end of this section compare total pipe lengths from water heaters to fixtures for both under-slab and above-slab piping systems.

Note that total pipe lengths from water heater to fixtures are usually greater in under-slab piping systems than in above-slab systems, as is discussed at the end of this section.

Multi-Story With Under-Slab Piping

Multi-story structures that have some under-slab piping have some features of under-slab piping and some features of above-slab piping. Depending on how the house is laid out and where piping is run, total piping lengths to fixtures may be shorter or longer than for single story structures. Where piping is run between floors instead of in the attic, piping is generally shorter than when run into the attic, because there is less vertical piping. Total pipe lengths from water heaters to fixtures in the case of multi-story buildings with under-slab piping are the sum of the under-slab piping lengths plus the above-slab piping lengths. These piping lengths can be estimated for a structure during the design phase by treating the above-slab pipe length as starting from the point where the piping exits the slab, and adjusting for additional vertical piping lengths as appropriate. For example when estimating piping lengths for a structure to be built, instead of a 3 foot length from water heater to attic, the appropriate pipe length from the slab to the floor between the first and second story would be estimated as approximately 8 feet (or whatever the appropriate ceiling height is for the structure) plus half the floor joist height. For estimating lengths of piping running between floors, an additional 8 feet of piping per floor (or other appropriate value if ceiling height is other than 8 feet) plus the floor joist and rough flooring heights would be a good assumption.

The examples discussed at the end of this section compare total pipe lengths for several different piping schemes.

Multi-Story With No Under-Slab Piping

In multi-story structures that do not have under-slab piping, total piping lengths from water heaters to fixtures can be estimated as described above for single-story structures if the piping is run in the floor between stories, except that for fixtures on the second story, the vertical lengths of piping in the walls are as discussed for under-slab piping.

If in multi-story piping some of the piping is run into the attic, for those branches an additional 8 feet per story (or other appropriate value based on actual ceiling height) plus the floor joist height and rough-flooring height should be added to the estimated pipe lengths, to account for the extra vertical pipe lengths in the walls.

The examples discussed at the end of this section compare total pipe lengths for several different piping systems.

Manifold/Parallel Pipe Distribution System

One of the sites visited used a hot water manifold distribution system with small-diameter PEX piping from the manifold to each fixture, as shown in figures 3-14 through 3-16. At this site, lines to sinks and showers were all nominal 3/8 inch diameter, except the main kitchen sink line, which was ½ inch. All other hot water lines went to tubs or the washing machine, and were ½ inch nominal diameter. All of the pipes were above-ground and none were insulated. At this site, the hot water distribution manifold was located a short distance away from the water heater and was connected to it by a nominal 3/4 inch diameter PEX line which was connected to several feet of 3/4 inch copper pipe near the water heater, yielding a total length of approximately 7 feet. All hot water piping, except a sink in the garage next to the water heater, and the clothes washing machine which was on the second floor immediately above the water heater, went from the distribution manifold up into the attic of the second story, and then back down again to each fixture, whether the fixture was on the first or second floor. All lines were continuous from the manifold to the fixture, with no fittings inside the walls or in the attic.

As seen in figures 3-14 through 3-16, piping lengths were significant at this site. However, pipe diameters were smaller than for trunk and branch or multiple main branch systems. Detailed piping length measurements were taken at this site, as summarized in table 3-4.

Table 3-4
Manifold/Parallel PEX Piping Length Measurements

NO.	FIXTURE	LINE SIZE (IN.)	LENGTH FROM MANIFOLD (FT.)
1	Not Connected		
2	M.Bath Lav.(sink)-1 (outside wall)	3/8	74.7
3	M. Bath Lav. (sink)-2	3/8	68.8
4	M. Bath Shower	3/8	57.0
5	Bath 2 Lav. (sink)-1 (upstairs)	3/8	52.1
6	Bath 2 Lav. (sink)-2 (upstairs)	3/8	52.8
7	Bath 3 Lav. (sink) (upstairs)	3/8	45.4
8	Bath 4 Lav. (sink) (downstairs)	3/8	62.6
9	Veggie Sink – Kitchen	3/8	98.9
10	Laundry Lav. (sink)	3/8	37.3
11-12	Not connected		
13	Garage Lav. (sink)	3/8	10
14	Clothes Washing Machine	½	11.1
15	Kitchen Sink + Dishwasher	½	96.8
16	M.Bath Tub	½	73.8

17	Bath 2 Tub (upstairs)	½	54.8
18	Bath 3 Tub (upstairs)	½	36.9
19	Bath 4 Tub/Shower (downstairs)	½	66.7
	TOTAL		900

One notable difference for the manifold distribution system compared to the other more conventional HWD systems seen, was that even with two side-by-side sinks in the same bathroom, each sink was given its own dedicated hot water line all the way back to the manifold. This means that use of hot water at one sink did not speed hot water delivery to the second sink.

An important observation with the manifold distribution system as implemented at this site is that, as seen in figure 3-16, the uninsulated hot and cold water lines were often bundled together and tightly bound with plastic wire ties. This means that there is probably substantial heat transfer between the hot and cold water lines, or between active and inactive hot water lines, during hot water draws, and moreover that piping cool-down rates will be even faster than they would be if the lines were not tightly bound together.

Another important observation with the manifold distribution system was the fact that the volume of the 3/4 inch line from the water heater to the distribution manifold represents a large percentage of the total volume between the water heater and the fixtures. This is because the lines to the fixtures have relatively small diameter, and hence volume. This large diameter line must be accounted for when computing piping water volumes. Additionally, the manifold itself was about 3 feet long, and also contained significant water volume – approximately the same as if it were a 3/4 inch diameter line.

Example Piping Length Comparisons

In order to better understand the differences in HWD piping lengths and "volumes-to-fixtures" between different types of piping systems in different applications, it is instructive to compare different piping schemes on a few example buildings. Appendix A details and compares an example single-story house with both under-slab and above-slab piping systems. It also does the same for an example two-story house with three different piping systems; one conventional trunk-and branch system with no under-slab piping, one trunk and branch system serving the second floor plus an under-slab system serving the first floor, and a manifold distribution system with separate small-diameter pipes to each fixture. The example house used in this case is similar to the actual house from where the manifold distribution system measurements were taken, and hence those measurements are used in the example. Some overall general observations from these examples are as follows:

Under-slab piping systems usually have significantly greater pipe volume between the water heater and the fixtures compared to above-slab piping systems, especially in single-story structures. One exception is when serving an "island" or "peninsula" fixture (one where the piping cannot be run in an adjacent wall directly to the fixture. In this case, an under-slab pipe run to the island fixture (e.g. an island kitchen sink) usually has less pipe volume than the above-slab piping case (which will have a below-slab pipe from a wall to the fixture). Another exception is for fixtures close to the water heater off the main trunk line. "Volume-to-fixture" can be somewhat higher for above-slab trunk-and branch systems in this case because the trunk-line close to the water heater is usually longer than the main trunk line leading to the multiple parallel branch lines of an under-slab piping system. In the single-story example of appendix A, under-slab piping "volumes-to-fixtures" range from 86% to 220% of the volumes of the above-slab trunk and branch system, with the ratio over 140% for most fixtures.

In two-story structures, much of the hot water piping is inherently above-slab. Piping runs to 1st floor island sinks such as the kitchen sink can benefit by being under-slab, but other piping runs generally won't unless they are close to the water heater and would otherwise branch off of a large diameter main trunk line. However, even though percentage differences may be fairly large for fixtures close to the water heater, total pipe volumes to those close fixtures are not normally large.

"Volume-to-fixture" for the manifold distribution system using small diameter pipe is significantly affected by the volume of the large diameter line from the water heater to the manifold, and by the volume of the manifold itself. The large diameter line going from water heater to manifold is often longer than the comparable main trunk line for a trunk-and-branch system, especially when the length of the manifold is also considered. Considering these extra volumes, the volumes from water heater to fixtures in the manifold system are sometimes substantially greater than with a normal trunk-and-branch system, and are sometimes significantly less. Manifold system lines that end up having more "volume-to-fixture" than trunk-and-branch systems are typically lines serving fixtures relatively close to the water heater. In the example two-story structure of appendix A, pipe "volume-to-fixture" for the manifold system ranged from 34% to 153% of the above-slab trunk-and branch system. Lines serving fixtures close to the water heater had typically 89% to 153% of the volume that the trunk-and-branch system would have had. Lines serving fixtures further away from the water heater typically had 34% to 65% of the volume that the trunk-and-branch system would have had. The overall average "volume-to-fixture" ratio for the manifold system compared to the above-ground trunk-and-branch system for this example was about 66%. The reader is cautioned, however, that pipe "volume-to-fixture" is not the only determining factor on water and energy waste. Cool-down rate is also important. Pipes that cool down slowly can maintain usable temperatures in the pipe between many draws, negating the need to purge cool water from the pipe in subsequent draws. The uninsulated small diameter pipes of a manifold distribution system will cool to below a usable temperature very quickly (2-3 minutes) and may therefore require more purging. More detailed analysis using realistic assumed draw patterns for each fixture will be necessary to draw conclusions about water and energy waste of manifold systems compared to other HWD system types.

FIGURE 3-1
PAX PIPING SHOWING ALUMINUM MIDDLE LAYER

FIGURE 3-2
PAX PIPING MAIN BRANCHES AT WATER HEATER

FIGURE 3-3
PAX PIPING AT HOUSE ENTRANCE – ONLY COPPER SHOWS OUTSIDE WALL

FIGURE 3-4
PAX PIPING WITH RED AND BLUE 6 MILL PLASTIC SHEATH AND KITCHEN LINE INSULATED

FIGURE 3-5
PAX PIPING: ONLY KITCHEN LINE INSULATED – SHOWS LONG LINES TO NEAR FIXTURES

FIGURE 3-6
LONG PAX PIPING – ONLY KITCHEN INSULATED, NO DIAGONAL PIPING

FIGURE 3-7
PAX ABOVE SLAB

FIGURE 3-8
PAX IN MULTI-FAMILY HOUSING, SAND BACKFILL IN PROGRESS

FIGURE 3-9
COPPER PIPING IN CUSTOM HOMES – MOSTLY INSULATED, BUT NOT ALL

FIGURE 3-10
COPPER PIPING IN CUSTOM HOMES – MOSTLY INSULATED, BUT NOT ALL
NOTE WET SAND AND STANDING WATER IN BACKGROUND

FIGURE 3-11
ABOVE-GROUND COPPER PIPING- NOTE POOR INSULATION AT ELBOWS

FIGURE 3-12
ABOVE-GROUND COPPER PIPING- MORE POOR INSULATION AT ELBOWS

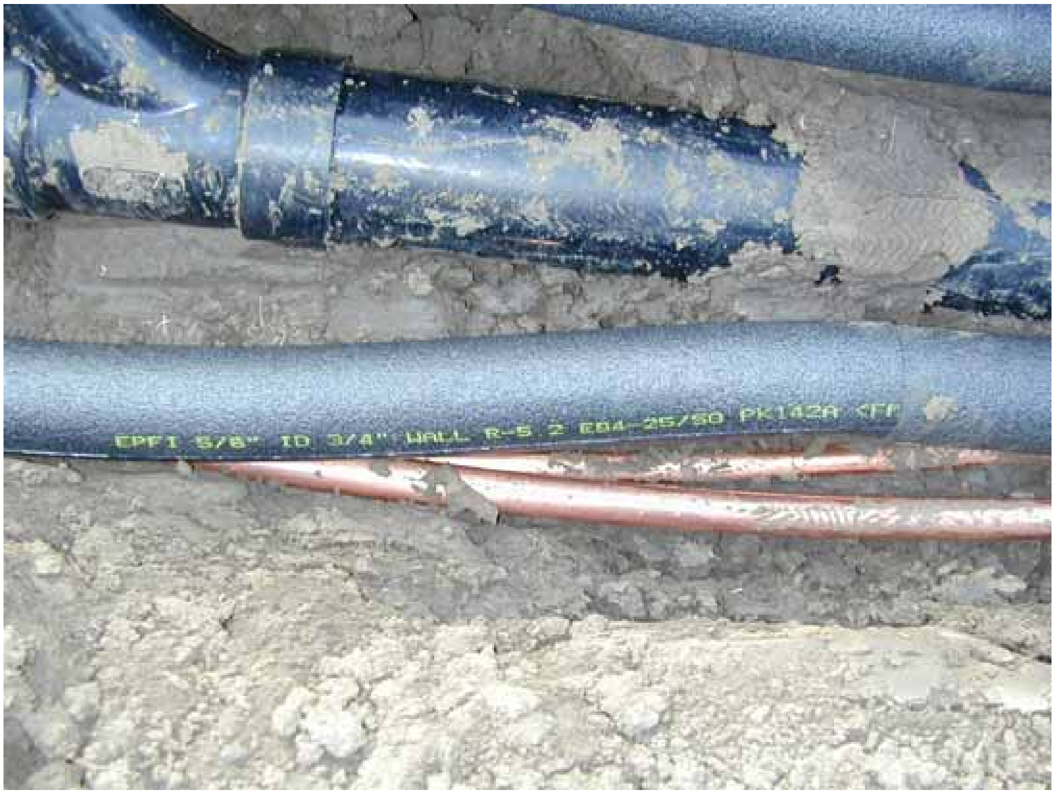


FIGURE 3-13
INSULATION NORMALLY MARKED – NOTE NO GLUE AT BUTT JOINT

FIGURE 3-14
MANIFOLD HOT WATER DISTRIBUTION SYSTEM

FIGURE 3-15
MANIFOLD DISTRIBUTION SYSTEM PIPING RUNS INTO ATTIC & BACK DOWN

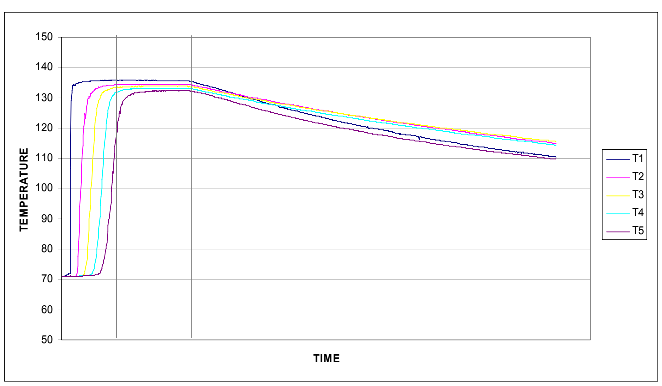
FIGURE 3-16
MANIFOLD DISTRIBUTION SYSTEM PIPING – NOTE HOT AND COLD LINES BOUND TOGETHER

4
TEST RESULTS – GENERAL OBSERVATIONS

A complete description of the laboratory test setup and procedures is given in Appendix B. As seen in figure 4-1, each test consisted of three phases. In the "delivery" phase, hot water traversed the pipe until usable hot water (105 F for test purposes) was available at the outlet. The "use" phase continued from the delivery phase and lasted from when 105 F water had reached the outlet until outlet temperature had remained relatively constant for at least 3 minutes. The "cool-down" phase began after the draw was terminated. The delivery-phase data was used to quantify time and water waste while waiting for "warm-enough" (105 F) water to arrive at fixtures. The use-phase was used to measure flowing pipe heat loss ($UA_{flowing}$) characteristics. The cool-down-phase was used to measure zero-flow pipe heat loss ($UA_{zero\text{-}flow}$) characteristics. The expressions for determining $UA_{flowing}$ and $UA_{zero\text{-}flow}$ are given in appendix B.

The tests conducted show that delivery-phase hot water flow in pipes is a highly transient, non-steady-state process, and that neither plug-flow nor fully-developed flow assumptions are valid. Plug-flow is defined as flow with a sharp hot-cold interface with no temperature degradation or mixing. Fully-developed flow is defined as flow where the boundary layer of the fluid in the pipe has reached its full steady-state thickness. In fact, fully-developed flow may never happen before a draw is terminated. Several distinct unanticipated flow regimes were observed that characterize delivery-phase flow in pipes where there is a hot/cold interface.

EXAMPLE TEMPERATURES VS TIME



DELIVERY PHASE USE PHASE COOL-DOWN PHASE

FIGURE 4-1
TEST PHASES: DELIVERY, USE, AND COOL-DOWN

Unanticipated Flow Regimes in Delivery-Phase Flow

Stratified Flow

One important difference of the delivery-phase flow in hot water distribution system piping compared to normal pipe flow is the fact that both hot and cold water are flowing simultaneously in the pipe. As hot water enters the pipe, cold water leaves the discharge end until that cold water is purged from the pipe. This leads to some unexpected behaviors under certain conditions in the vicinity of the hot/cold interface in the pipe. In particular, in rigid copper pipe, flow stratification was observed to occur, where hot water traveled a further distance down the top side of the pipe than it did on the bottom side. This phenomenon is explained more fully in the next few paragraphs. Stratification also happened in the PAX piping at low flow rates, but there was another phenomenon that overshadowed it – see the subsection below on slip-flow.

Flow stratification, where hot water floats on top of colder water due to density and viscosity differences, increases water and energy waste while waiting for "warm-enough" water to arrive at fixtures. This is because the stratification causes an increased surface area for heat transfer from the hot water to the cold water in the pipe. This heat transfer occurs both directly between the hot and cold water, and indirectly through the highly conductive pipe wall from the hot water to the cold water. This heat transfer lowers the temperature of some of the hot water that enters the pipe to below a usable temperature, causing an increase in water waste (and hence energy waste). The stratification appeared to persist for much or all of the first straight section of the test piping, but flow had usually become fully mixed before the end of the second straight section. Stratification heads (region where hotter water is above colder water in the pipe) at least 6 feet long were observed in some situations. The stratification appeared to occur at low flow rate conditions where the flow of the cold water in the pipe was laminar as discussed below.

Dimensionless Reynolds number (Re) is a measure of flow turbulence. Higher Re means flow is more turbulent, having a large number of flow eddies that cause mixing. At lower Re, flow becomes laminar, having no flow eddies. For flow in pipes

$Re = \rho V D_i / \mu$

Where:
ρ = density
μ = viscosity
V = flow velocity
D_i = pipe inner diameter

This can be stated for flow in circular pipes as a function of volumetric flow rate as

$Re = 4\,\rho(GPM)/(\mu\,\pi D_i)$

Where:
GPM = Volumetric flow rate in units that make Re dimensionless

For water in the range of 40 F to 160 F, density varies from 62.4 to 61.0 lbs/ft^3, but viscosity (a measure of shear resistance in the fluid) varies from 3.81 to 0.98 lbm/hr ft. Thus we can see that for any given pipe diameter, Re varies strongly as a function of temperature because of changes in viscosity. Cold water is much more viscous than hot water, and therefore resists movement more strongly than the hot water. Re is also a strong function of the volumetric flow rate and the pipe diameter. For water flow in circular pipes, flow is laminar below about Re = 2300.

Test results in the ¾ and ½ inch rigid copper piping showed that stratification of flow occurred approximately whenever the cold water Re was laminar. As shown in Table 4-1, this occurred at low flow rates, typically less than about 0.9 gpm in the ¾ inch rigid copper pipe and 0.6 gpm in the ½ inch, and then only when the water initially in the pipe was fairly cold.

Table 4-1
Reynolds Numbers in Pipe: Tc = 40 F, Th = 160 F

Flow Rate (gpm)	**¾ Rigid Cu**		**½ Rigid Cu**		**¾ PAX**	
	Cold	Hot	Cold	Hot	Cold	Hot
0.5	**1277**	4851	**1840**	6988	**1244**	4725
1.0	2810	9703	3680	13975	2488	9450
2.0	5109	19405	7359	27951	4976	18900

Note that flow rates in the 0.5 to 0.6 gpm range occur frequently in hot water piping systems. For example, most sinks demonstrate maximum flow rates in the 1.0 to 1.5 gpm range, due to the use of aerators that restrict flow. If the user desires 105 F water at the tap, but the hot water reaching the tap is at 135 F, then cold water is mixed with hot water at the fixture to deliver the desired flow rate of 105 F water. Flow rates actually used for hand washing and hand dish washing are often in the range of 0.5 gpm to 1.0 gpm. From an energy balance on the mixing of hot and cold water to obtain the 105 F water, it can be shown that the ratio of hot water flow rate to total flow rate is:

$(m_{hot})/(m_{total}) = (105\text{ F} - T_{cold})/(T_{hot}-T_{cold})$

At 135 F hot water and 75 F cold water temperatures this ratio becomes:

$(m_{hot})/(m_{total}) = (105 – 75)/(135-75) = 0.50$

Hence, at a desired total end use flow rate of 0.5 to 1.0 gpm, the actual hot water flow rate would be 0.25 to 0.5 gpm. Even under conditions of low hot and cold water temperatures, such as 115 F and 40 F, the hot water flow ratio is $(m_{hot})/(m_{total}) = (105 – 40)/(115-40) = 0.87$, resulting in actual hot water flow rates of 0.4 to 0.87 gpm. These results mean that stratification is a regularly occurring phenomenon in real piping systems.

Axial Heat Conduction Wave

Another phenomenon that appeared to play at least a slight role in increasing water and energy waste during the delivery phase of hot water flow was axial heat conduction in copper pipe. Often this effect is negligible, but sometimes it is not. Under many conditions, due to the relatively high thermal mass of copper, the rate of axial heat conduction in the pipe is less than the flow velocity of the water in the pipe. However, it was discovered analytically that at low flow rates, such as less than 0.5 gpm in ¾ inch rigid copper pipe, the propagation rate of axial heat conduction was actually faster than the water flow velocity in a small region just ahead of the hot water flow head. This means that at low flow rates, there is effectively a “bow wave” of heat conducting down the pipe just ahead of the hot water. This “bow wave” of heat represents an increased surface area for heat transfer from the hot water to the cold water, through the pipe wall. Since flow velocity decreases as pipe diameter increases at a given volumetric flow rate, this axial heat conduction bow wave phenomenon is expected to play a larger role in large diameter conductive pipes experiencing low flow rates. When flow velocity increases, this phenomenon ceases to be important whenever the hot water flow travels faster than the heat conduction wave.

Slip-flow

Another extremely important flow phenomenon observed was what we term “slip-flow”. It appeared that above certain flow velocities, which varied with pipe size, type, and temperature conditions, heat transfer and mixing between hot and cold water in the pipe were substantially less than would be predicted by standard fully-developed flow theory. It appeared that friction between the water and the pipe wall was substantially less than one would normally see in fully-developed flow, resulting in reduced heat transfer from the water to the surrounding ambient, and also in reduced turbulence and mixing of hot and cold water in the pipe. In short, it appeared that water flow was slipping through the pipe, shearing off the boundary layer so as to prevent the development of turbulent eddies.

Slip-flow was observed in all the pipes tested during this phase of the project, but was most pronounced in the very smooth PAX piping. Not only was the PAX piping extremely smooth inside, but it also lacked elbows and other fittings that could trigger turbulence. Additionally, water in the plastic wall material had lower surface adhesion than in the copper pipe, causing the water to adhere less to the wall. This lower surface adhesion caused the cold water to resist motion less in the plastic pipe than in the copper.

The slip-flow phenomenon was an important aspect of the non-steady-state behavior of delivery-phase flow in the piping, and could not be predicted based on Re, flow velocity, or other properties of just the water alone. The pipe wall smoothness and material also play a role. The effects of this slip-flow were most pronounced in the entrance regions of the piping test sections, but sometimes persisted through the entire length of the test section and sometimes for an extended period. Due to its nature, slip-flow is an unsteady process and flows could go in- and out-of slip-flow during a test under some conditions. At longer pipe lengths and longer flow times, flow usually gradually ceased to be slip-flow. However, in some tests, slip-flow persisted over the entire test section length even for extended periods – fully-developed flow never developed!

Slip-flow is desirable because it reduces heat transfer and mixing that would degrade temperature of the hot water as it passes through the pipe, hence reducing water and energy waste and speeding delivery of hot water.

Mixing and Other Pipe Internal Effects On Delivery-Phase Flow

In any piping system, if the temperature of the water in the pipe at the beginning of the draw is lower than useful, that water must be wasted to drain (or recirculated back to the water heater, which we will discuss separately) while waiting for “warm-enough” water to arrive at fixtures. The amount of water wasted to drain is, at a minimum, equal to the volume of water within the pipe. In reality, however, the temperature of the water leaving the water heater is degraded by a variety of mechanisms on its way to the fixture, which necessitates that an amount of water even greater than pipe volume be wasted to drain before “hot-enough” water arrives at fixtures. There are several important mechanisms that cause a degradation of the temperature of water leaving the water heater as it travels through hot water distribution system piping to fixtures. In non-recirculated systems, those mechanisms are:

1. Heat transfer from the hot water to heat up the mass of the pipe and insulation (the slip-flow phenomenon described above reduces this effect on any particle of water traveling through the pipe).

2. Direct conductive heat transfer between the hot and cold water within the pipe (the stratification phenomenon described above exacerbates this heat transfer)

3. Indirect conductive heat transfer between the hot and cold water within the pipe, through heat conduction in the pipe wall (both the stratification and axial heat conduction phenomena described above exacerbate this heat transfer).

4. Mixing of hot and cold water within the pipe due to flow turbulence (the slip-flow phenomenon described above reduces this effect).

5. Heat loss to surrounding ambient (see separate discussion below).

In non-recirculated HWD systems, water in the pipe will be fairly cold during the first draw of the day, having cooled-off over night, whether insulated or not. For subsequent draws, the water in the pipe may or may not be at a usable temperature, depending on the surrounding ambient conditions, the amount of insulation on the pipe, size (and hence thermal mass) of the pipe, and amount of time since the last draw. The amount that the pipe has cooled off between draws is hence an important factor in determining water and energy waste and is discussed separately later in this section.

The first four mechanisms described above are not directly dependent on surrounding ambient conditions, but are rather indirectly related because they depend to some extent on the initial temperature of the water and pipe, which are impacted by ambient conditions. These four mechanisms increase water and energy waste because they require the dumping of some partially heated water to drain, and because they require that a greater amount of the remaining partially hot water be mixed with cold water to obtain a desired amount of water at a desired end-use temperature than if the heat loss had not occurred.

Heat Transfer to Ambient Effects

The tests performed under this project allow quantification of the impacts of piping heat loss on water and energy waste. Piping heat loss impacts water and energy waste both directly and indirectly, through the following mechanisms:

1. Under conditions where heat energy is lost from water traveling through the piping, but the water still arrives at the fixture at a usable temperature (such as in prolonged steady-state flow), energy use is increased by one or both of two mechanisms:

    a. A drop of temperature from the water heater to the fixture causes use of more hot water to mix with cold water to achieve a desired end-use temperature and volume, compared to if there were no piping heat loss (i.e. cases where the water heater setpoint temperature is higher than the minimum necessary, and flow rate of hot water can be increased)

    b. A drop of temperature from the water heater to the fixture requires that the setpoint temperature of the water heater be increased so that the same amount of hot water is used to mix with cold water to achieve a desired end-use temperature and volume, as would be experienced if there were no piping heat loss (i.e. cases where the water heater is set to the lowest possible delivery temperature or the flow rate to the fixture cannot be increased due to inadequate pressure or flow restrictors). Note, however, that an increase in tank setpoint temperature incurs an energy loss from the water heater tank that occurs continuously 24 hrs/day, 365 days/yr – it does not just impact piping energy loss when draws are occurring. Stated differently, this means that water heaters that are located closer to fixtures can be set to lower temperatures than those that are further away, and hence both piping heat loss and tank heat loss can be reduced. Tank heat loss is hence an important consideration and must be considered in HWD system energy loss comparisons.

2. Under conditions where so much heat energy is lost from the water traveling through the piping that the water does not arrive at the fixture at a usable temperature (such as at very low flow rates and/or with very long pipes, or pipes traveling through high-heat transfer environments such as bare pipe in wet or damp soil), water and energy waste are increased because of both the direct energy loss, as in number 1 above, and because the remaining warm water is not useful and must be dumped to drain, taking its energy content down the drain also.

    a. In some cases at low flow rates and/or long pipe lengths, or in high heat-transfer environments, hot-enough water may never arrive at fixtures, requiring that 100% of the water and energy content of that water is wasted unless the tank setpoint temperature is increased. As noted above, an increase in tank setpoint temperature incurs an energy loss from the water heater tank that occurs continuously 365 days/yr, and must be considered when comparing energy waste of different HWD layout options.

    b. During the delivery phase of a hot water draw, when hot water is traveling for the first time to the fixture, heat transfer to the surrounding ambient can reduce the temperature of some of the warm water that reaches the fixture to below a usable temperature. This increases both water and energy waste in a manner similar to that described above for pipe internal mixing and other effects.

Pipe Cool-Down Effects

One of the largest water and energy wastes caused by HWD piping is the cool-down of hot water left in piping after draws cease. The magnitude of this water and energy waste depends on several factors:

1. Volume of piping between water heater and fixture. The greater this volume, the higher the water and energy waste.

2. HWD piping layout impacts both piping length and volume, and the extent to which draws from some fixtures preheat some of the water that will later need to go to other fixtures, thus reducing the effective volume of water that will need to be wasted.

3. The amount of pipe insulation

4. The environment surrounding the pipe

5. The size and hence thermal mass of the pipe and the water contained within it (affects volume of water wasted and cool-down time)

6. Time spacing between draws

Volume of Piping

If draws are far enough apart that the HWD piping has cooled to below a usable temperature between the draws, the cool water in the piping must be wasted to drain and displaced with hot water from the water heater. This hot water then looses its energy content to ambient as the pipe cools down again. Using smaller diameter pipes, if allowed by plumbing codes, reduces pipe volume, and hence water and energy waste.

More importantly, however, locating water using fixtures closer to hot water sources can reduce both the length and diameter of the HWD system piping required. The shorter length and smaller diameter reduce the amount of water that must be wasted to drain while waiting for warm-enough water to arrive at fixtures. It also reduces tank heat losses by enabling use of lower delivery temperatures from the water heater to the piping. It is often possible to achieve substantial water and energy savings merely by more centrally locating the water heater relative to the fixtures. Sometimes using more than one water heater to reduce total HWD piping length is also beneficial from a time, water, energy, and even a first cost perspective.

HWD Piping Layout and Preheating From Use of Other Fixtures

As noted above, modifications to HWD piping layout can have a dramatic effect on water and energy waste. Additionally, in some piping systems, such as trunk-and-branch type systems, use of water at one fixture may bring hot water past or nearer to another fixture than would be the case if the first fixture had not been used. This impacts the amount of water and energy that will be wasted when using hot water from the subsequent fixtures. Practical applications of this knowledge would suggest putting fixtures that are likely to be used in similar time-frames on the same HWD circuits. Doing so would result in draws more closely spaced in time, with reduced pipe cool-down between draws and hence reduced water and energy wasted down the drain.

Amount of Pipe Insulation

The laboratory testing of this project shows that use of pipe insulation can dramatically reduce both the rate of heat loss from piping, and the cool-down rate. If pipes pass through high heat transfer environments, such as through concrete, wet, or damp areas, or very cold areas, pipe insulation is highly desirable. The test results discussed in detail later in this report show that extreme amounts of insulation are not required, and that benefits of insulation are non-linear. That is to say, the first small thickness of insulation reduces heat loss a lot, while greater thicknesses provide diminishing benefits.

Use of insulation is extremely important in HWD systems that use recirculation loops (RL), because piping is hot much of the time with such systems, and there is typically almost twice as much piping compared to non-RL systems.

Additionally, pipe insulation has the very important effect of reducing the cool-down rate of water in pipes after flows have stopped. In many cases, pipe insulation can reduce the cool-down rate enough such that water in the piping remains above a useful temperature between draws, compared to bare piping which cools down much quicker. (See pipe cool-down rate tables in section 8 of this report.) Pipe insulation thus provides both direct energy and water waste reduction benefits from reducing heat loss, and sometimes even larger leveraged benefits by reducing the need to dump partially heated water to drain.

Environment

Heat transfer from pipes to ambient depends strongly on both the temperature difference between the hot water in the pipe and the surrounding ambient, and on the heat transfer characteristics of that environment. For pipes in still air, heat transfer coefficients on the outside of the pipe are relatively low, and hence heat transfer rates per foot are normally relatively low (note however that total heat loss rates can still be high if piping is long). If the surrounding air is moving, heat transfer rates to the air will be substantially higher than for still air. Similarly, if uninsulated pipes are run in high-thermal-mass environments, such as buried in sand, soil, or concrete, or are touching dense materials such as sheet-rock, the effective thermal mass of the piping system will be substantially increased, which will increase water and energy waste because more hot water will be needed to heat up the surrounding environment to allow the pipe to heat up. Worse yet, if the pipes are run in wet or damp environments, such as buried in wet sand, heat loss rates from piping (especially bare piping) can be more than an order of magnitude higher than for the in-air environment (see water bath test results in section 11 of this report). At a minimum, HWD pipes should probably be insulated if they will be run in cold environments, or will be in contact with high-heat transfer environments such as touching or buried in wet sand, concrete, sheet-rock, or other water pipes. However, in some buried environments, especially if dry, the burial materials may effectively act as insulation once the mass of the material is heated up. More research is recommended to quantify the impacts of these environments under different draw scenarios.

Pipe Size and Thermal Mass

Larger diameter pipes hold more water per foot than do smaller diameter pipes, requiring that once those pipes have cooled off to below a usable temperature, a larger amount of water must be wasted to drain before again obtaining usably hot water. Moreover, the energy lost when that larger amount of water cools to below a usable temperature is greater than for smaller diameter pipes. Countering these effects to some extent, however, is the fact that since larger pipes hold more water per foot and hence have greater thermal mass than smaller pipes, they have lower surface area/volume ratios and hence they take longer to cool down than do smaller pipes. This means that water in larger pipes remains above a usable temperature for a longer period between draws compared to smaller pipes. This could result in less water and energy waste to drain for larger diameter pipes compared to smaller diameter ones under some draw spacing scenarios. More analysis of piping system performance under a variety of draw scenarios is recommended to better understand conditions under which increased pipe diameter is beneficial vs detrimental.

Time Spacing Between Draws

Time spacing between draws is important in determining magnitude of water and energy waste from non-recirculated piping systems. This is because once draws cease, pipes begin to cool off. If the next draw on that piping circuit occurs beyond a time by which the pipe has cooled below a useful temperature, then any remaining energy in warm water in the pipe must be wasted to drain in order to again obtain usably hot water. The water and energy waste are then basically the full energy content of hot water refilling the pipe, plus an extra amount due to mixing within the pipe and other heat transfer effects as described above during the delivery phase. However, if draws are spaced closely enough such that the water in the piping has not yet cooled to below a usable temperature, then “hot enough” water is immediately available without dumping water and energy to drain. In this latter case, the only energy loss is due to the extra amount of “hot enough” water that must be mixed with cold water in order to obtain desired amounts of warm water at a desired end-use temperature, compared to if the heat loss had not occurred.

From a HWD system design standpoint, this all means that pipes should be insulated, avoid running pipes in high-heat transfer environments, try to minimize pipe lengths and volumes, and try to put end uses that will occur in similar time frames on the same piping runs if possible. From an operational perspective, it also means timing end uses on the same circuits to occur close enough to each other in time that the piping will stay “hot enough to use” between draws. However, the latter must be done with recognition that there may be adverse impacts on water heater sizing (and hence heat loss) if too many large hot water draws are scheduled too close together in time. Studying the cost-effectiveness of these measures is beyond the scope of the current work.

Recirculation-Loop HWD Systems

While the laboratory testing performed under this Phase I contract did not test hot water recirculation-loop (RL) systems, much of the data on piping heat loss rates and delivery-phase water and energy waste is still directly applicable to analysis of water and energy waste in RL systems.

The measured pipe UA factors given later in this report can be directly applied to calculate the heat loss off of RL piping. In fact, this piping heat loss analysis is simpler in RL systems than in non-RL systems, because there are far fewer pipe cool-down calculations to be performed, and time between draws is relatively unimportant if the RL is kept hot anyway. It is important to remember that RL systems have both supply and return hot water piping, and heat loss must be calculated off of both sets of pipes.

The delivery-phase water and energy waste analyses determined from the laboratory test results also directly apply to the analysis of hot water delivery in branch lines off the main RL loop line. In this case, the RL serves as the "water heater" for the branch line, and the total RL system water and energy waste are the sum of the wastes from the main RL loop piping plus the branch lines.

Some special considerations apply when attempting to predict water and energy waste from "on-demand" RL systems, where the hot water is circulated in the HWD system only in response to a need for hot water. Activation of the RL system circulating pumps is achieved by a variety of different schemes, each of which behaves differently from an energy and water waste perspective. Examples include pump activation by use of manual push-buttons, occupancy sensors, flow sensors, and combinations of the above. Other activation methods probably also exist. Additionally, RL pump termination is achieved by a variety of different schemes, each of which behave differently from an energy and water waste perspective. Examples include pump deactivation by use of return-temperature sensors, local pipe temperature sensors, simple pump run-time controllers, and combinations of the above. Other pump deactivation methods probably also exist. Note that in some systems, the temperature sensors used to turn the pump off may also prevent the pump from turning back on, and hence function as part of the control system for both activating and deactivating the pump. For example, a temperature sensor that turns the pump off at a specified temperature (e.g. 95 F), may also prevent the pump from activating until the pipe temperature drops below that value.

Another important consideration in RL systems, whether or not they are of the "on-demand" type, is that typically more of the piping system is heated than is just needed to serve the fixture demanding the draw of hot water. How much piping is heated (and is hence loosing heat) in RL systems depends on the specific system design and control strategies, and how the controls really work.

It is not permissible to simply assume that RL systems have no water waste. At a minimum RL systems have some water waste from their branch lines. In RL systems, additional water and energy waste may occur because the controls may not activate or deactivate in an ideal fashion. For example, some local pipe temperature activation/deactivation controls are set to not let the pump activate if pipe temperatures are below a useful temperature (say 105 F), but above some fairly warm temperature (say 90 F). This may prevent the RL pump from coming on, requiring dumping of all the water in the RL loop before “hot enough” water can arrive at fixtures. More work is required to study exactly how RL systems are controlled and how they really work. Such work is beyond the scope of the currently reported effort.

General AF/PV Results

As discussed in Appendix B, it was learned that the best way to present information on water and energy waste during the delivery-phase of flow was to provide the ratio of actual flow (AF) gallons needed to deliver usefully hot water (we define as 105 F) divided by pipe volume (PV) as a function of relevant variables. It was found that dependence on temperature conditions could be reasonably represented by the ratio (Thot-105 F)/(Thot – Tpipe initial). We refer to this ratio as the temperature drop ratio (TDR) because conceptually it represents the ratio of the allowable temperature drop while still obtaining water at a useful temperature (105 F) divided by the maximum possible temperature drop that could occur in the flow.

TDR = $(T_{hot}\text{-105 F})/(T_{hot} – T_{pipe\ initial})$

Where:

T_{hot} = Hot water temperature entering the piping system

$T_{pipe\ initial}$ = Initial temperature of water in the pipe

105 F = Defined minimum useful temperature

Note that TDR varies between 0 and 1.0 for situations of interest. If TDR is mathematically greater than 1.0, it means the pipe is already hotter than 105 F, and there is no “down-the-drain” water or energy waste since hot water is instantly available. High TDR values (values above 0.5) usually correspond to conditions of high entering hot water temperature and high initial pipe temperature. Low TDR values (values below 0.25) usually occur at low hot water temperatures and low initial pipe temperatures.

The AF/PV ratio varies between 0 and infinity, but is usually between 1.0 and 2.0. At TDR ≥ 1.0, AF/PV goes to 0 because the pipe is already hot and hot water is instantly delivered. At any TDR < 1.0, AF/PV is not less than 1.0. Note, however, that strictly speaking TDR is a function of pipe length because heat loss to ambient can degrade the temperature of the hot water as it flows through the pipe. This means, for example, that an entering TDR might be 1.0 because the pipe is already at 105 F, yielding an AF/PV ratio of 0 in the entrance region of the pipe, but if the hot water temperature is fairly low and the pipe is in a high heat transfer environment, heat loss from the pipe would degrade the temperature of the water in the pipe and cause AF/PV ratio to go rapidly to infinity over a short length. The values of TDR presented in this report are all "initial TDR", or TDR based on test section entering hot water temperature. However, when modeling HWD system performance, continuously calculating TDR as a function of pipe length would allow accurate simulation while simultaneously accounting for pipe heat transfer to ambient.

Higher AF/PV ratios translate into greater water and energy waste while waiting for "hot enough" water to arrive at fixtures. In general higher AF/PV ratios occur with less-well insulated (or uninsulated) pipe, pipe in high heat transfer environments, low TDR ratios, and low flow rates. Variation of AF/PV ratio with pipe length depends on flow regime and heat transfer environment, as will be discussed more fully in the various detailed results sections. An AF/PF ratio of 1.0 indicates that flow in the pipe was close to perfect plug flow with essentially a sharp hot/cold temperature front moving in the pipe and no measurable degradation in hot water temperature as flow went to the fixture. A value of infinity means so much heat energy was lost to the environment that water at or above 105 F could not be delivered.

In laboratory testing AF/PV ratios of infinity were observed on several occasions. These were cases where the water flow rate was low (e.g. 0.5 gpm), entering hot water temperature was low (e.g. 115 F), the heat loss environment was severe (cold air – e.g. 50 F, no pipe insulation), and the piping was long (e.g. 100+ ft). In such cases enough heat was lost from the hot water in the pipe to ambient that at longer pipe lengths the temperature dropped below 105 F. It should be noted, however, that for all pipe lengths and insulation levels, AF/PV ratio goes to infinity at very low flow rates. Also, for all flow rates, AF/PV goes to infinity at very long pipe lengths. In most cases these low flow rates and long pipe lengths are outside of what are seen in normal every day conditions. However, sometimes these conditions do occur in normal practice, especially if piping is uninsulated in a high-heat-loss environment. In these cases, the user would normally increase the tank setpoint temperature to compensate, since increasing the flow rate may not be possible because of fixture flow-rate limitations. The "critical length" beyond which 105 F water cannot be delivered at any given flow rate, temperature, pipe insulation, and other conditions, can be calculated using pipe UA values. Examples of calculated critical length for some conditions are given in section 8 of this report.

Behavior of AF/PV ratio as a function of relevant parameters is easiest to understand by plotting AF/PV ratio vs both flow rate and pipe length. We therefore normally present the AF/PV results in two plots. One is AF/PV ratio vs flow rate, cross-plotted against TDR and pipe length. The other is AF/PV ratio vs pipe length, cross-plotted against TDR and flow rate. Figures 4-2 and 4-3 show generic examples of these plots, and the various kinds of behaviors that were observed depending on flow regime. Different flow regimes are marked as A, B, C, D, and E in the figures.

Region A of figures 4-2 and 4-3 is the region where heat transfer between water in the pipe and the surrounding ambient is important. Region A corresponds to low flow rate and/or long pipe lengths, typically under low TDR (TDR in the range of 0 to 0.3), and generally to uninsulated pipe. The low TDRs here are typically the result of low hot water temperatures. However, given enough pipe length, all AF/PF ratios will go to infinity, insulated or not, regardless of flow rate. Similarly, all AF/PF ratios will go to infinity at extremely low flow rates.

Region B of figures 4-2 and 4-3 is a region where two effects are important. One is stratification of flow in the pipe, with the entering hot water flowing on top of the colder water that was originally in the pipe. The other is axial conduction along the pipe from hot water to cold water. Both of these effects result in increased-length mixed temperature regions that increase wasted gallons. Region B corresponds to low flow-rate, low TDR conditions (TDR in the range of 0 to 0.4). The low TDRs here are typically the result of low initial pipe water temperatures, causing laminar flow in the cold water initially in the pipe at the low flow rates. The greater viscosity of the cold water restricts its motion initially while the less viscous hot water flows more freely on the top side of the pipe due to density differences. It is probable that the axial conduction effect, which occurs because velocity of heat conduction in the pipe wall in the vicinity of the temperature wave exceeds flow velocity, will be different in pipes of different materials. Note that these effects occur mainly in the pipe entrance regions, and become difficult to see at longer pipe length as general mixing continues to spread the length of the hot/cold interface zone.

Region C of figures 4-2 and 4-3 is a region where extremely high entrance region mixing may be prevalent. We are unsure whether or not this is a real effect or just an artifact of measuring difficulties at high flow rates in the entrance regions of small diameter pipes. It occurred in only a couple of tests. If real, it appears to correspond to high flow-rate, low TDR conditions in the entrance region of small-diameter pipes. The low TDRs in this case are caused by low hot water temperature. We will investigate this behavior more thoroughly if we see it in future tests.

Region D of figures 4-2 and 4-3 appears to be characterized by a transition of flow regime to slip-flow under conditions of high flow rate and high TDR (TDR above 0.4). The high TDRs here are the result of both high hot water temperatures and high initial pipe temperatures. Pipe insulation appears to improve the chances that flow will become slip-flow, by keeping pipe inner surface temperatures higher than for bare pipe. The higher temperatures reduce water viscosity, thus reducing shear resistance of the flow. When flow trips to slip-flow, the flow behaves much like adiabatic plug-flow, with very sharp well-defined temperature gradients in the flow and small incremental wasted gallons above pipe volume. In this flow regime AF/PF ratio has been observed at 1.0 in several tests in the pipe entrance region. As can be seen in the detailed results sections and appendices, slip-flow appears to happen more readily in the very smooth PAX piping, which also lacks conventional elbows and other fittings that could trigger turbulence. It is as yet unknown whether or not similar behavior would be seen in other types of pipe having fewer fittings.

Region E: The majority of test results for AF/PF fall somewhere in the center regions of figures 4-2 and 4-3. The AF/PF ratios in the majority of the tests are influenced mostly by mixing of hot and cold water as the flow proceeds and heat transfer between the hot water and the pipe thermal mass, both of which degrade some of the hot water to below usable temperatures. Typically encountered AF/PF ratios appear to be in the range of 1.1 to 2.0

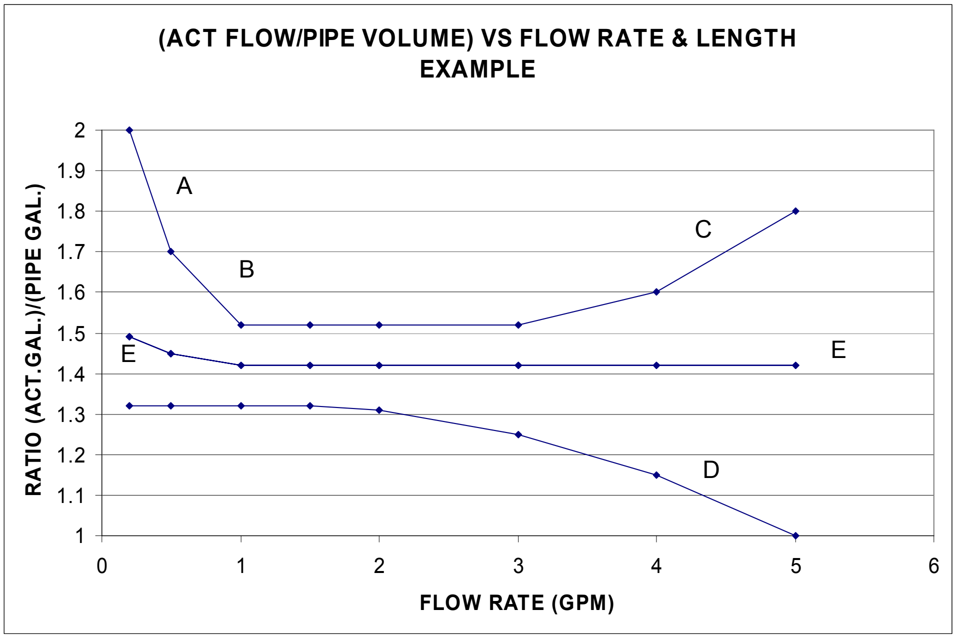


FIGURE 4-2
EXAMPLE BEHAVIORS
AF/PV RATIO OF WASTED GALLONS VS FLOW RATE

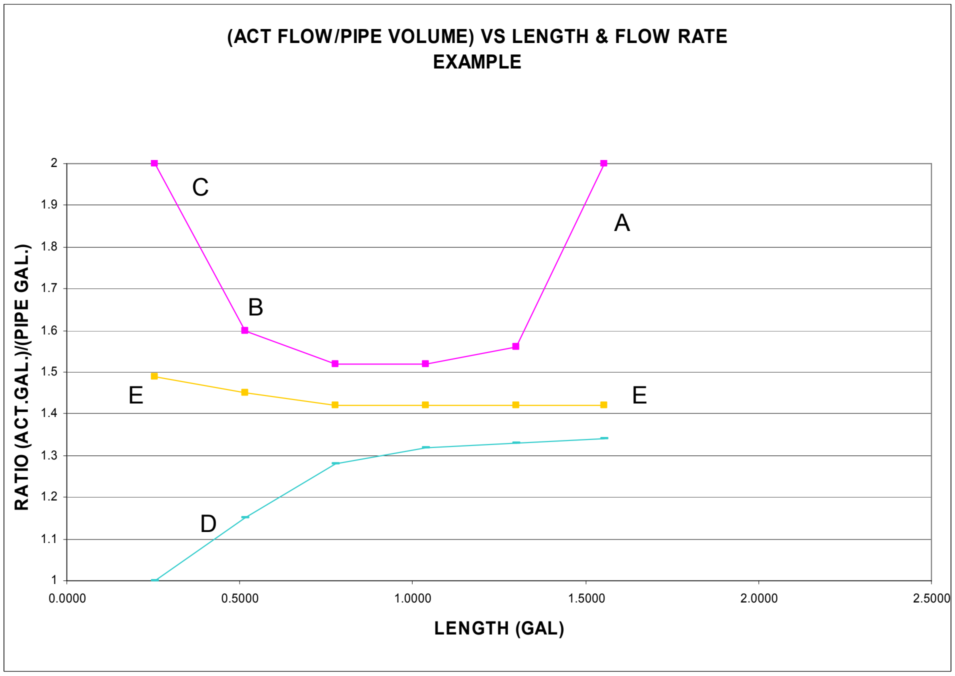


FIGURE 4-3
EXAMPLE BEHAVIORS
AF/PV RATIO OF WASTED GALLONS VS PIPE LENGTH

5 TEST RESULTS – HORIZONTAL ¾ INCH RIGID COPPER PIPE IN AIR

Test and calculational procedures and the flow and temperature test matrix for the horizontal in-air tests are described in Appendix B and will not be repeated here. Detailed presentation of test data, complete with photographs of the test setups and results summary tables for all pipe lengths, flow rates, and TDR cases tested are given in Appendix C. Only summary results are presented here.

The ¾ inch rigid copper piping tests were done on a test section having four straight lengths each slightly over 20 feet in length, plus three U-bends. Immersion thermocouples were inserted in the center of each U-bend, as shown in figures B-2, B-3, C-1 and C-2. Total test section length was approximately 86.3 feet.

Piping Heat Loss UA Values

Appendix C shows detailed results for UA values determined during testing on ¾ inch rigid copper pipe in air with no insulation, ½ inch thick foam insulation (R-2.9 hr ft^2 F/Btu) and ¾ inch thick foam insulation (R-4.7). Those results show that UA is a function of both the temperature difference between the pipe and the surrounding air and the flow rate. In general UA values are higher at higher temperature differences, meaning that pipe heat loss is greater at higher temperature differences in a non-linear manner. For practical engineering calculations, however, we can curve-fit the highest UA values observed (generally the higher temperature difference cases), and use those values for most calculations as if they were independent of temperature difference. Figure 5-1 show scatter-plots of all the data points for the various UA values vs flow rate for all the tests, plus curve-fit lines through the highest UA values vs flow rate for each insulation level. Note that variation of UA with temperature difference is most pronounced for bare pipe, and is much smaller for insulated pipes. This is probably because the actual pipe surface temperature for heat transfer varies more with flow rate in the bare pipe case than do the insulation external surface temperatures. Note especially that the $UA_{zero\text{-}flow}$ for bare pipe varies substantially, and this value should theoretically not be a function of temperature difference between the water in the pipe and the air, because each cool-down test sees continuously dropping pipe temperature, and hence continuously dropping temperature difference. One therefore has substantial latitude in what value one uses to curve-fit the $UA_{zero\text{-}flow}$ data point for the bare pipe case.

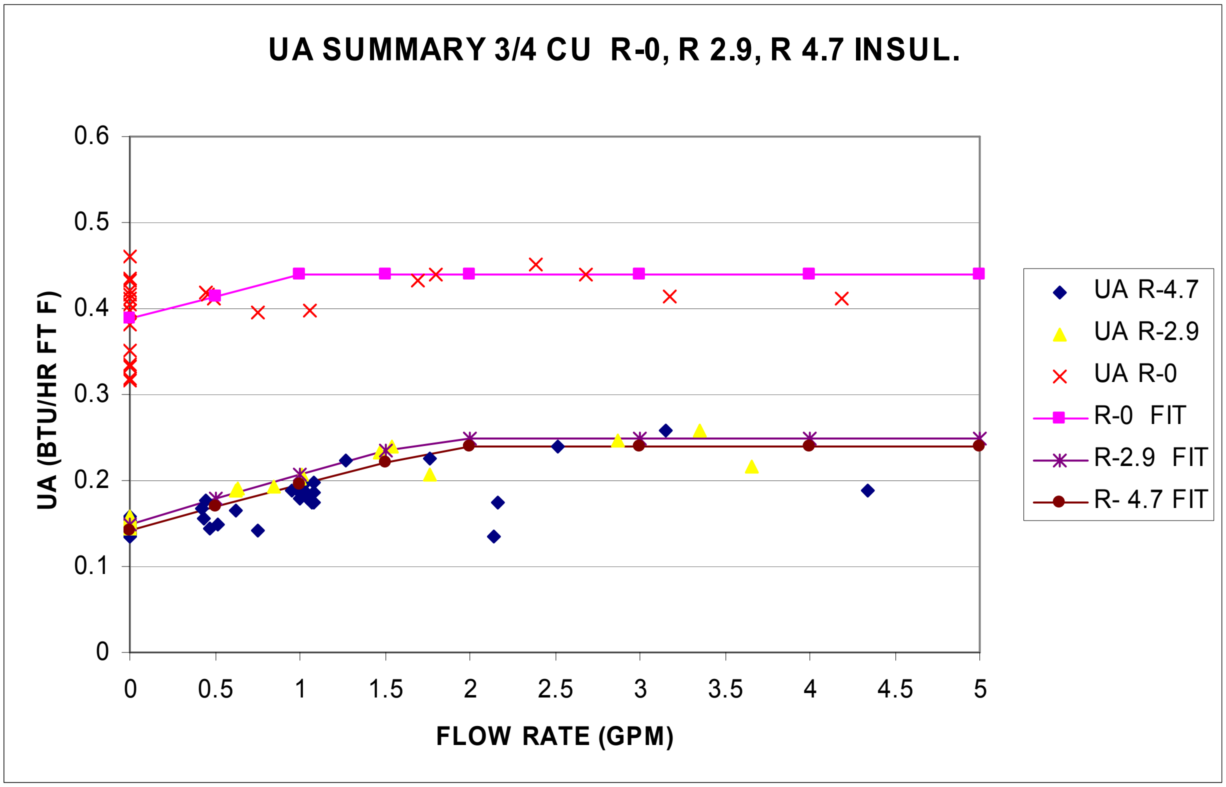


Figure 5-1
Measured Pipe Heat Loss UA Values for ¾ inch Rigid Copper Pipe

Figure 5-1 shows that for ¾ in rigid copper pipe, UA values are a function of water flow rate, but they level out and become constant at flow rates above about 2.0 gpm. It also shows that UA values at zero flow are generally somewhat lower than for cases where there is flow

Two important observations regarding the value of insulation can be drawn. The first is that adding even a relatively thin layer of insulation (1/2 inch thick foam) cuts pipe heat loss rate almost in half. The second is that using thicker insulation provides only slightly better performance. That is to say, the first small amount of pipe insulation provides a large reduction in pipe heat loss, and increasing insulation thickness provides highly non-linearly increasing benefits, with percentage improvement decreasing rapidly with increasing insulation levels.

Delivery-Phase AF/PV Ratio

Section 4 contains a discussion of the general types of behaviors of AF/PV ratio that were seen during testing, so that discussion will not be repeated here. Appendix C gives complete data summary tables for all valid tests. This section focuses on presenting quantitative summary results.

¾ Inch Bare Rigid Copper Pipe AF/PV Results

Figures 5-2 and 5-3 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the ¾ inch bare rigid copper pipe tests. Figure 5-2 shows results for a pipe length of 21.7 ft (the end of the first straight section), corresponding to 0.5454 gallons pipe volume. Figure 5-3 shows results for a pipe length of 86.3 ft – the end of the entire test section, corresponding to 2.169 gallons pipe volume. Figures 5-4 and 5-5 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.5 and 4.0 gpm respectively.

The individual plots in figures 5-2 through 5-5 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.15 – 0.20) and lower lines corresponding to the highest TDRs tested (nominally 0.50 – 0.60). The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

Comparing figures 5-2 through 5-5, we see that for this pipe configuration AF/PV ratios were highest in the entrance region of the test section, especially at low TDR. At higher TDR the entrance region effects largely disappeared. Additionally we see that at low TDR and low flow rates, AF/PV was substantially higher than at higher TDR and/or higher flow rates. The combination of short pipe lengths, low TDR, and low flow rates causes among the highest AF/PV ratios for this pipe configuration, and corresponds to region B as discussed in Figures 4-2 and 4-3, where flow stratification and pipe axial heat conduction have a detrimental affect on AF/PV ratio. We also note that at low TDR, AF/PV improves with pipe length. This means that the detrimental effects of stratification and pipe axial heat conduction that occur in the pipe entrance region are largely negated if the pipe is long enough.

In figure 5-4, we can also see that AF/PV ratio begins to increase with increasing pipe length after having reached a minimum, for low TDR and low flow rates. This corresponds to region A as discussed in Figures 4-2 and 4-3, where pipe heat transfer to the surrounding ambient is particularly detrimental. If the pipe had been slightly longer, AF/PV ratio would have gone to infinity (see the ½ inch copper and ¾ inch PAX results).

Figures 5-6 and 5-7 show all AF/PV results from all tests on bare ¾ inch rigid copper piping, plotted on the same plots. In figure 5-6 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs and longer pipe lengths. In figure 5-7 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix C allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 5-6 and 5-7 that for ¾ inch bare rigid copper piping, AF/PV ratios normally range from a low of around 1.1 to a high of around 2.0, with most in the range of 1.2 to 1.5.

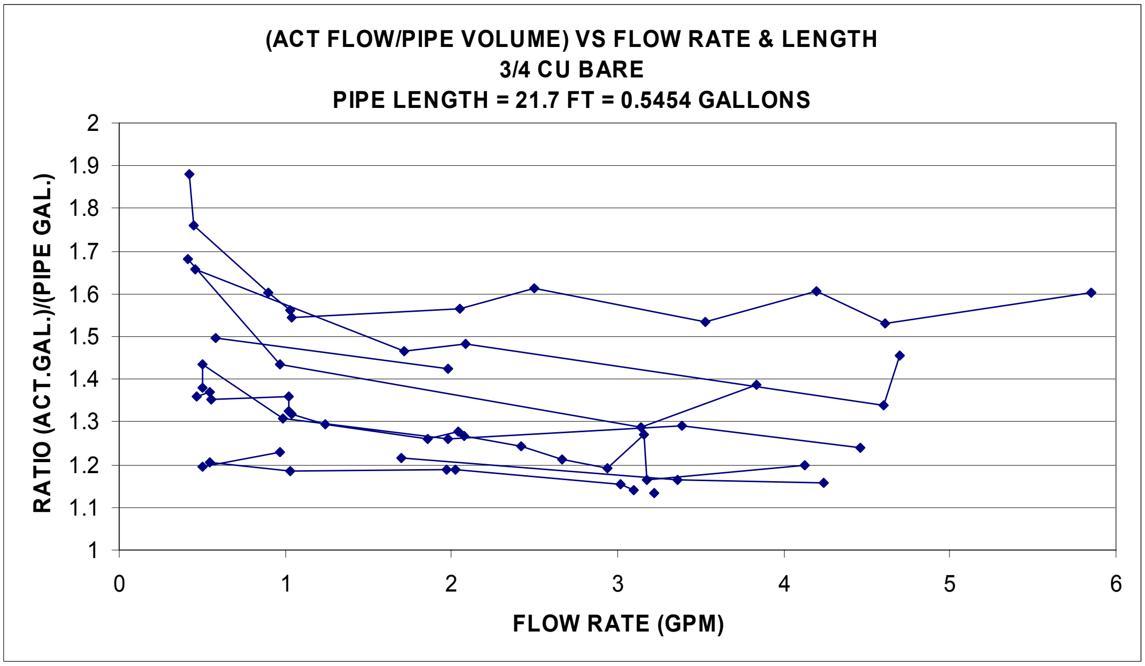


Figure 5-2
Measured AF/PV Ratio vs Flow Rate and TDR For Bare ¾ Inch Rigid Copper Pipe – 21.7 Ft Pipe Length (Upper Plot TDR = 0.153 – 0.194, Lower Plot TDR = 0.50 – 0.60)

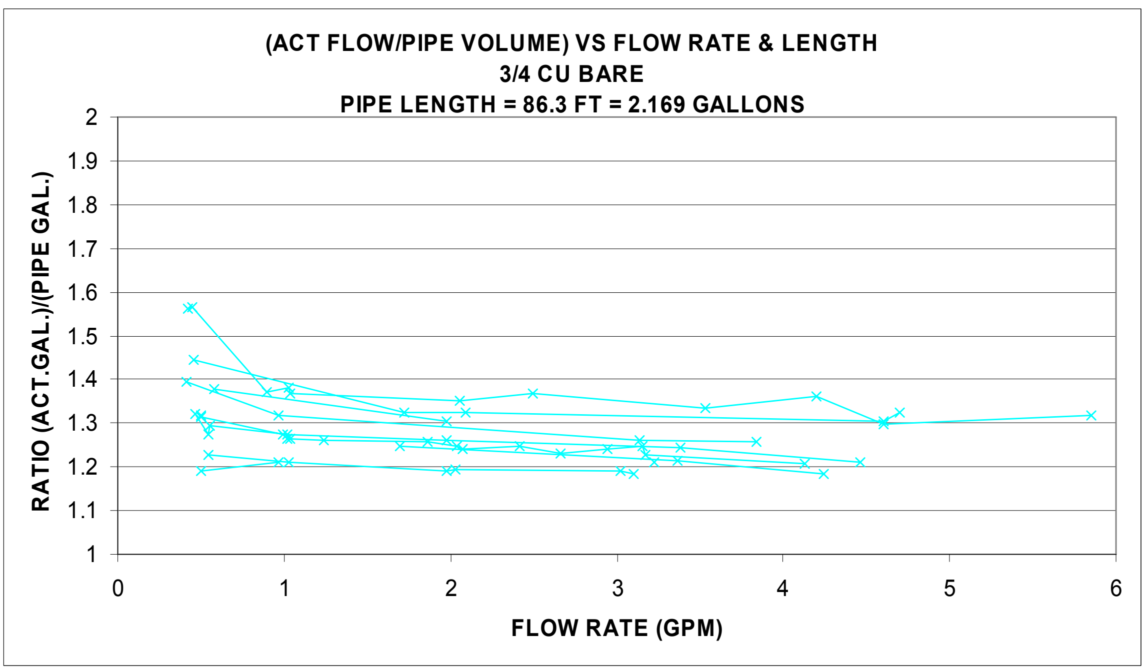


Figure 5-3
Measured AF/PV Ratio vs Flow Rate and TDR For Bare ¾ Inch Rigid Copper Pipe – 86.3 Ft Pipe Length (Upper Plot TDR = 0.153 – 0.194, Lower Plot TDR = 0.50 – 0.60)

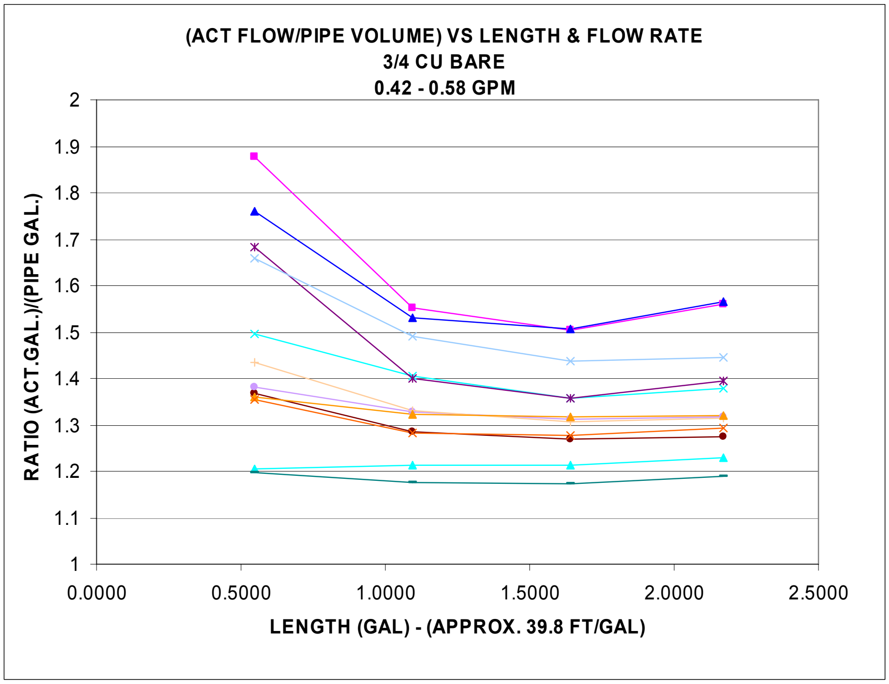


Figure 5-4
Measured AF/PV Ratio vs Pipe Length and TDR For Bare ¾ Inch Rigid Copper Pipe – 0.42 - 0.58 GPM Flow Rate (Upper Plot TDR = 0.178, Lower Plot TDR = 0.618)

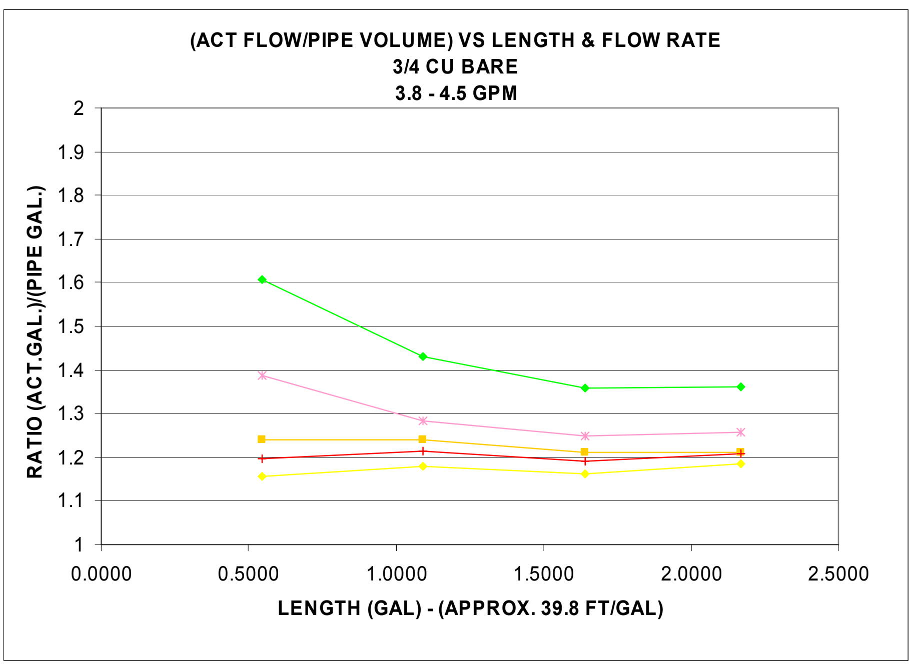


Figure 5-5
Measured AF/PV Ratio vs Pipe Length and TDR For Bare ¾ Inch Rigid Copper Pipe – 3.8 - 4.5 GPM Flow Rate (Upper Plot TDR = 0.164, Lower Plot TDR = 0.497)

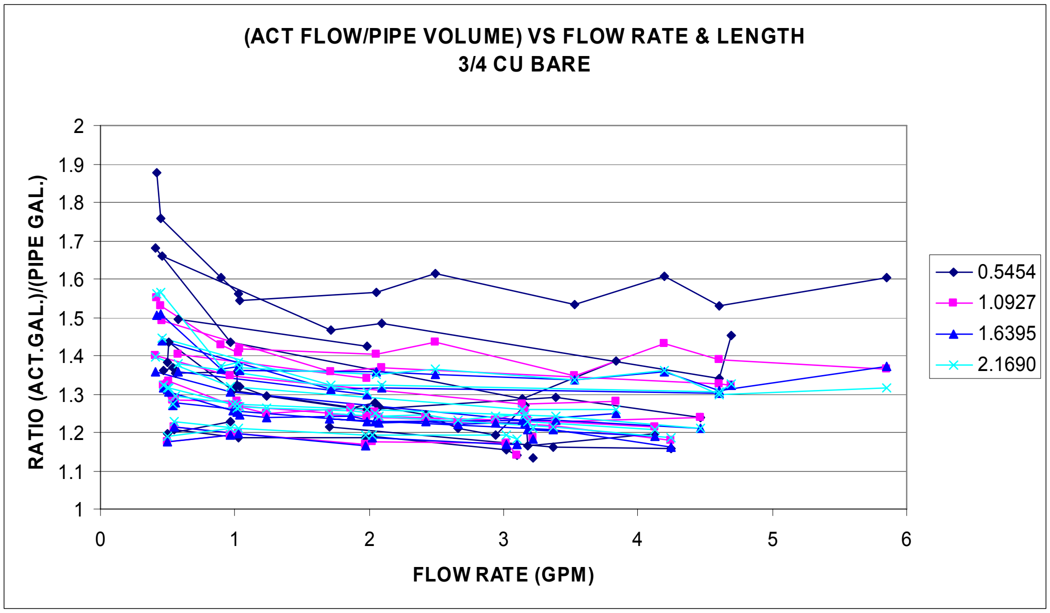


Figure 5-6
Measured AF/PV Ratio vs Flow Rate, Length, and TDR For Bare ¾ Inch Rigid Copper Pipe – All Pipe Lengths and TDRs (Legend is pipe length in gallons, where 39.78 ft = 1 gallon)

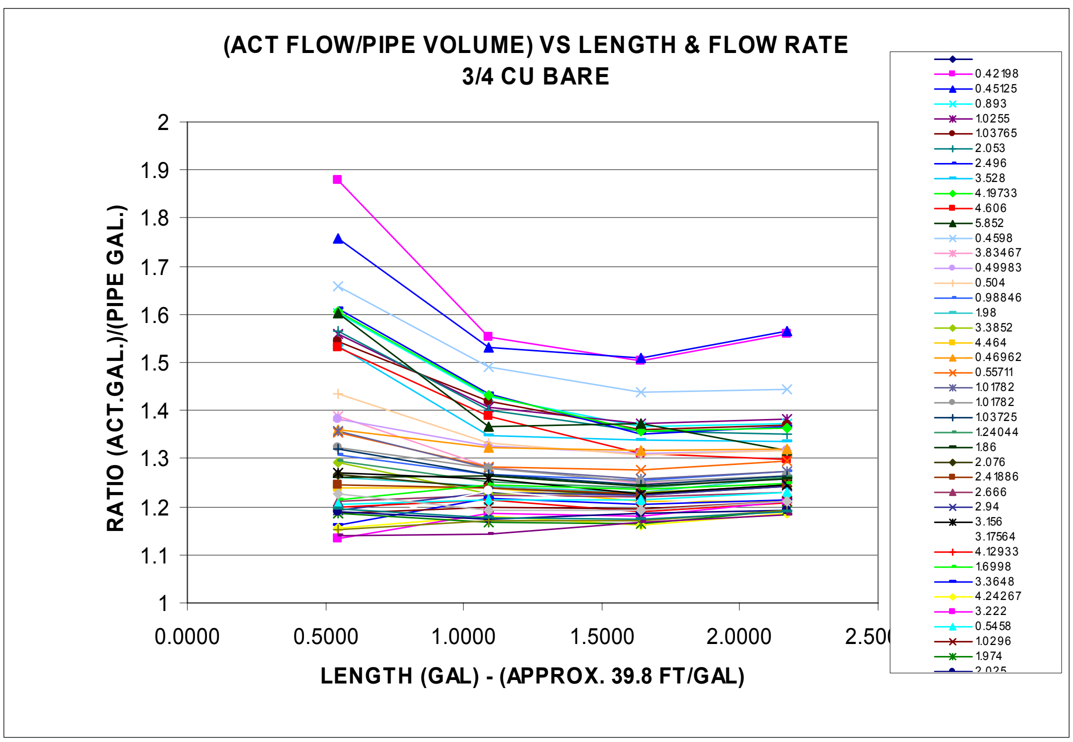


Figure 5-7
Measured AF/PV Ratio vs Pipe Length, Flow Rate, and TDR For Bare ¾ Inch Rigid Copper Pipe – All Flow Rates and TDRs (Legend is flow rate in gpm.)

¾ Inch Rigid Copper Pipe With ½ Inch Thick Foam Insulation AF/PV Results

The foam pipe insulation used for these tests was of the same type most commonly seen in the field site visits - black closed-cell polyethylene or polyolefin foam. Properties for these two materials are similar. R-values were printed on the insulation, like was observed at the field sites. The typical thermal conductivity listed by various manufacturers for these types of pipe insulation was around k = 0.02 Btu/hr ft F. The R-values ranged as shown in table 3-2 for the different foam thicknesses and different pipe sizes, because R-value is based on the outer diameter of the foam. The above listed thermal conductivity can be used in standard heat transfer calculations for multi-layer cylindrical objects (given in most introductory heat transfer textbooks) to calculate the R-value for insulation of a given thickness applied to pipe of a given diameter. The R-value of the ½ inch thick foam on the ¾ inch nominal copper pipe size (approximately 7/8 inch OD) was R-2.9.hr ft^2F/Btu

Figures 5-8 and 5-9 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the tests on ¾ inch rigid copper pipe with R-2.9 insulation. Figure 5-8 shows results for a pipe length of 21.7 ft (end of first straight section), corresponding to 0.5454 gallons pipe volume. Figure 5-9 shows results for a pipe length of 86.3 ft (end of entire test section, corresponding to 2.169 gallons pipe volume. Figures 5-10 and 5-11 show AF/PV ratios vs pipe length, cross-plotted against TDR for 0.5 and 4.0 gpm flow rates respectively.

The individual plots in figures 5-8 through 5-11 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.39 – 0.44) and lower lines corresponding to the highest TDRs tested (nominally 0.45 – 0.47). The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

The ¾ rigid copper with R-2.9 insulation tests showed results similar to the uninsulated tests with several exceptions. First, at lower flow rates, AF/PV ratios were slightly lower for the R-2.9 tests compared to the bare tests, at similar TDRs. This suggests that heat transfer to ambient was affecting the AF/PV ratio of the low-flow-rate bare-pipe tests even at short pipe lengths. Also, region A of figures 4-2 and 4-3, where heat transfer to ambient is a significant effect, did not exist with the insulated pipe. The most significant difference in behavior was observed at high flow rates and high TDR. As seen in the lower plot on figures 5-10 and 5-11, at high flow rates and short pipe lengths, AF/PV ratio was significantly lower than for the bare pipe tests. After seeing similar results in later pipe tests it was recognized that this lower AF/PV ratio was occurring because flow in the entrance region of the insulated pipe at high TDR and high flow rates was entering the slip-flow regime. This was identified as region D in figures 4-2 and 4-3. It appears that the slip-flow degraded to normally developing flow before the end of the first straight length of the test section, and that by the end of the entire test section the slip-flow effect had largely been overcome by normal mixing and water-to-pipe-wall heat transfer.

Figures 5-12 and 5-13 show all AF/PV results from all tests on ¾ inch rigid copper piping with R-2.9 insulation, plotted on the same plots. In figure 5-12 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs and longer pipe lengths.

In figure 5-13 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix C allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 5-12 and 5-13 that for ¾ inch rigid copper piping with R-2.9 insulation AF/PV ratios mostly behaved similarly to the bare pipe case, with the normal AF/PV ratios observed being above 1.2, except when slip-flow occurred.

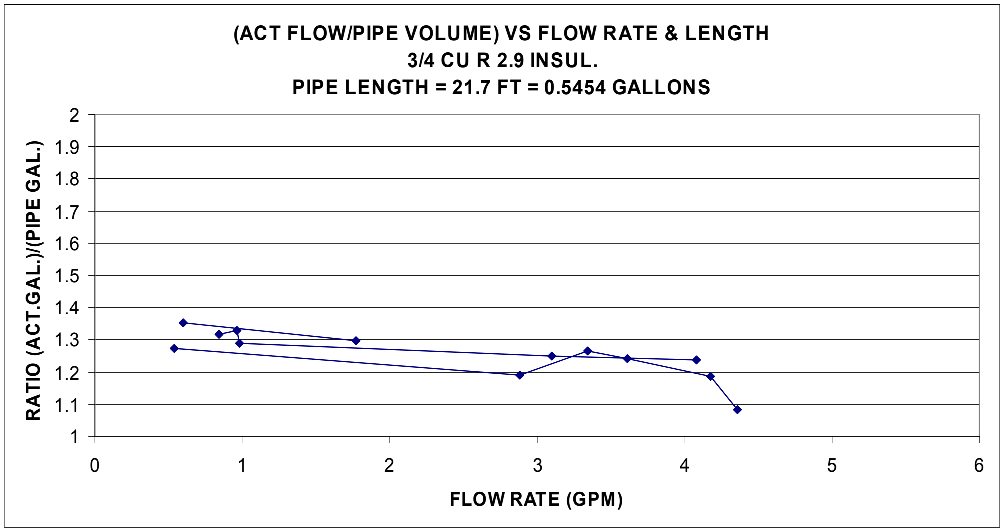


Figure 5-8
Measured AF/PV Ratio vs Flow Rate and TDR For ¾ Inch Rigid Copper Pipe With R-2.9 Insulation– 21.7 Ft Length (Upper Plot TDR = 0.389 – 0.396, Lower Plot TDR = 0.450 – 0.473)

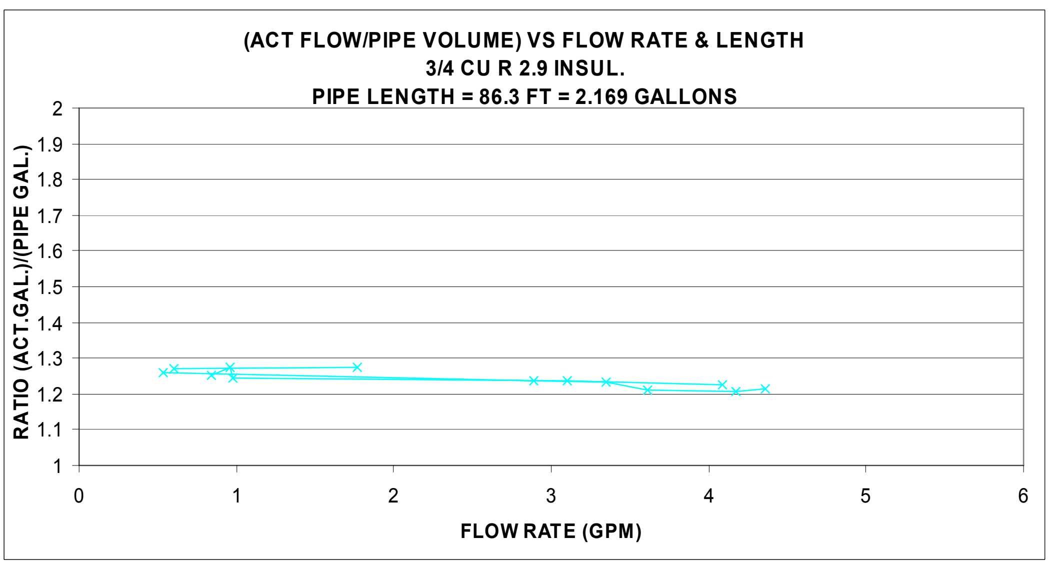


Figure 5-9
Measured AF/PV Ratio vs Flow Rate and TDR For ¾ Inch Rigid Copper Pipe With R-2.9 Insulation– 86.3 Ft Length (Upper Plot TDR = 0.389 – 0.396, Lower Plot TDR = 0.450 – 0.473)

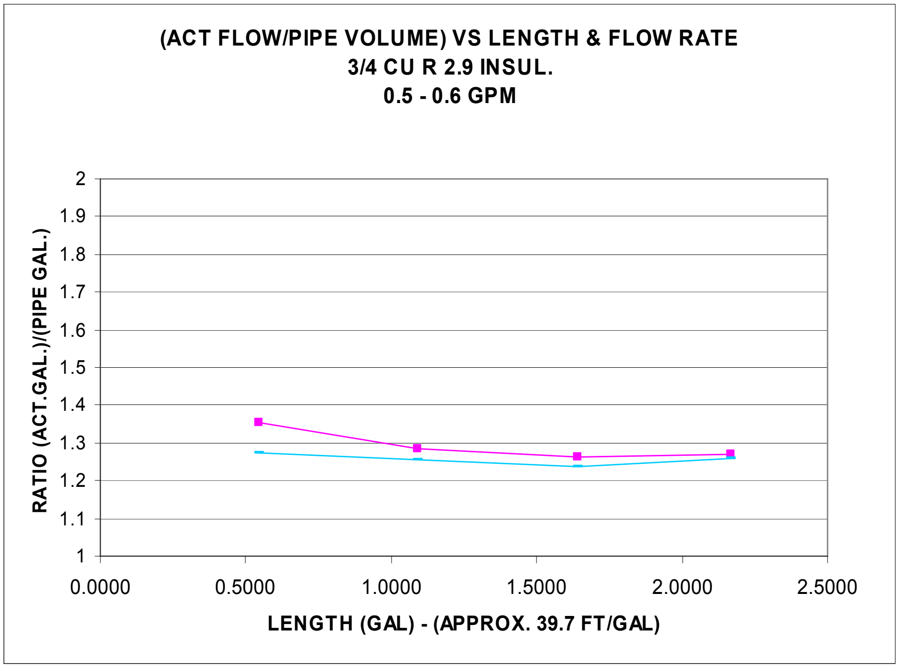


Figure 5-10
Measured AF/PV Ratio vs Pipe Length and TDR For ¾ Inch Rigid Copper Pipe With R-2.9 Insulation– 0.5 – 0.6 GPM (Upper Plot TDR = 0.389, Lower Plot TDR = 0.450)

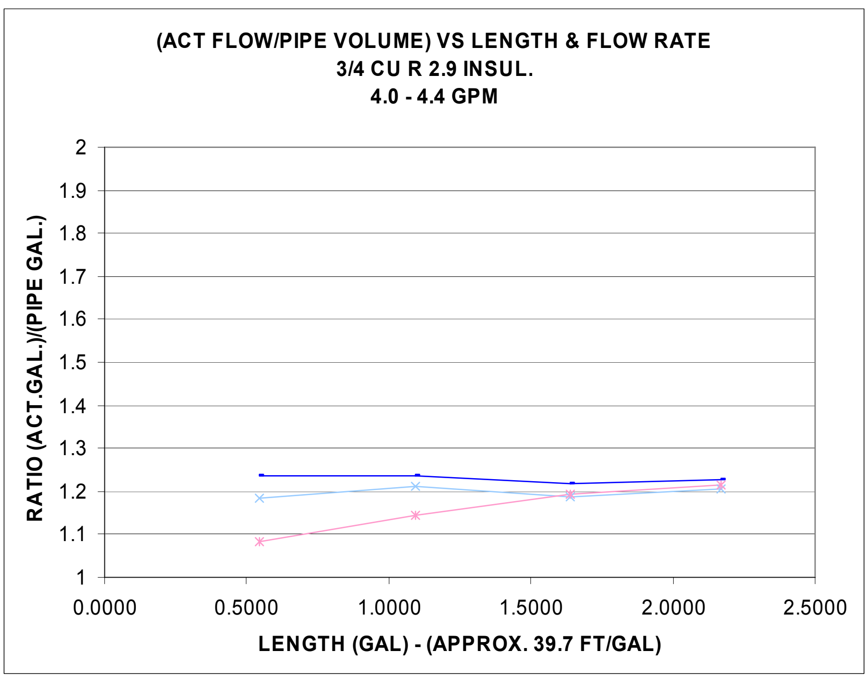


Figure 5-11
Measured AF/PV Ratio vs Pipe Length and TDR For ¾ Inch Rigid Copper Pipe With R-2.9 Insulation– 4.0 – 4.5 GPM (Upper Plot TDR = 0.442, Lower Plot TDR = 0.473)

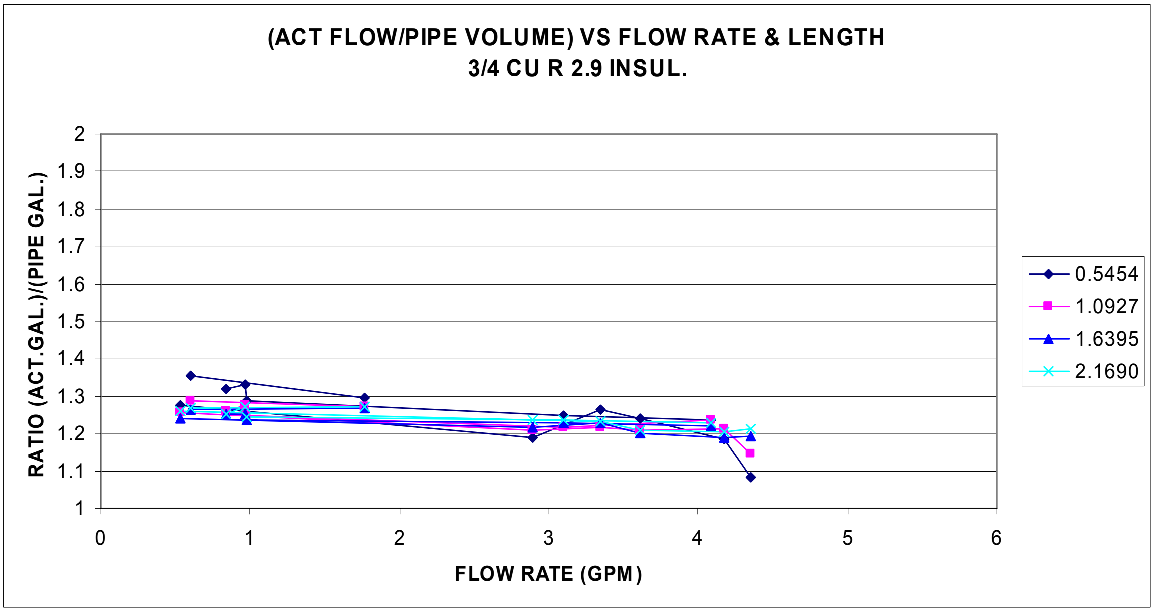


Figure 5-12
Measured AF/PV Ratio vs Flow Rate, Length & TDR For ¾ Inch Rigid Cu Pipe With R-2.9 Insulation– All Lengths and TDRs (Legend is length in gallons, where 39.78 ft. = 1 gallon.)

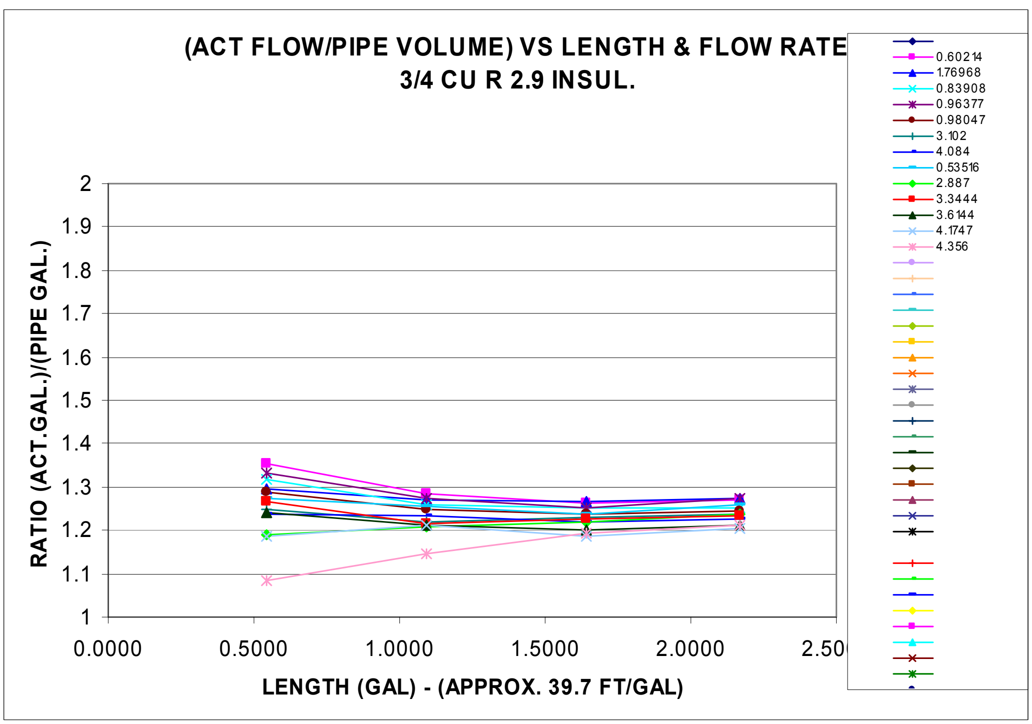


Figure 5-13
Measured AF/PV Ratio vs Pipe Length, Flow Rate, and TDR For ¾ Inch Rigid Copper Pipe With R-2.9 Insulation– All Flow Rates and TDRs (Legend is flow rate in gpm.)

¾ Inch Rigid Copper Pipe With 3/4 Inch Thick Foam Insulation AF/PV Results

The R-value of the 3/4 inch thick foam on the ¾ inch nominal copper pipe size (approximately 7/8 inch OD) was R-4.7 hr ft^{2}F/Btu.

Figures 5-14 and 5-15 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the tests on ¾ inch rigid copper pipe with R-4.7 insulation. Figure 5-14 shows results for a pipe length of 21.7 ft (the end of the first straight section), corresponding to 0.5454 gallons pipe volume. Figure 5-15 shows results for a pipe length of 86.3 ft – the end of the entire test section, corresponding to 2.169 gallons pipe volume. Figures 5-16 and 5-17 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.45 and 4.5 gpm respectively.

The individual plots in figures 5-14 through 5-17 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.178) and lower lines corresponding to the highest TDRs tested (nominally 0.575). The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

The ¾ rigid copper with R-4.7 insulation tests showed results similar to the uninsulated and R-2.9 insulated tests with several exceptions. First, like for the R-2.9 tests, at lower flow rates, AF/PV ratios were slightly lower for the R-4.7 tests compared to the bare tests, at similar TDRs. This suggests that heat transfer to ambient was affecting the AF/PV ratio of the low-flow-rate bare-pipe tests even at short pipe lengths, but that there was little difference between the impacts of R-2.9 vs R-4.7 insulation on AF/PV ratio. Like for the R-2.9 tests, there was no region A behavior, where heat transfer to ambient significantly affected AF/PV results. The most significant difference in behavior was observed at high flow rates and high TDR. As seen in the lower plot on figure 5-17, at high flow rates, AF/PV ratio was significantly lower than for the bare pipe tests, and the difference persisted over a greater pipe length in the R-4.7 tests compared to the R-2.9 tests. After seeing similar results in later pipe tests it was recognized that this lower AF/PV ratio was occurring because flow in the entrance region of the insulated pipe at high TDR and high flow rates was entering the slip-flow regime. This was identified as region D in figures 4-2 and 4-3. Unlike the R-2.9 tests, however, it appears that the slip-flow effect continued throughout the entire test section. It appears that at the slightly higher TDR of this test (TDR = .575) compared to the highest TDR tested for the R-2.9 test (TDR = .473), flow in the pipe remained on the borderline of going in and out of slip-flow vs normally developing flow for the entire 86.3 foot test section.

Figures 5-18 and 5-19 show all AF/PV results from all tests on ¾ inch rigid copper piping with R-4.7 insulation, plotted on the same plots. In figure 5-18 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs and longer pipe lengths. In figure 5-19 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix C allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 5-18 and 5-19 that for ¾ inch rigid copper piping with R-4.7 insulation

AF/PV ratios mostly behaved similarly to the bare pipe case, with the normal AF/PV ratios observed being between 1.2 and 2.0, except when the slip-flow occurred.

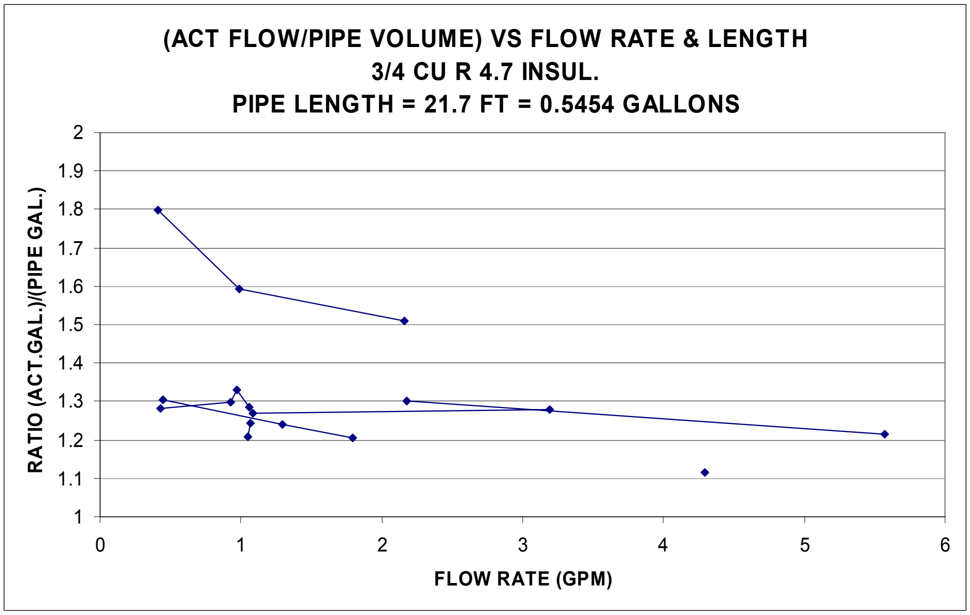


Figure 5-14
Measured AF/PV Ratio vs Flow Rate and TDR For ¾ Inch Rigid Copper Pipe With R-4.7 Insulation– 21.7 Ft Length (Upper Plot TDR = 0.178 – 0.199, Lower Plot TDR = 0.450 – 0.575)

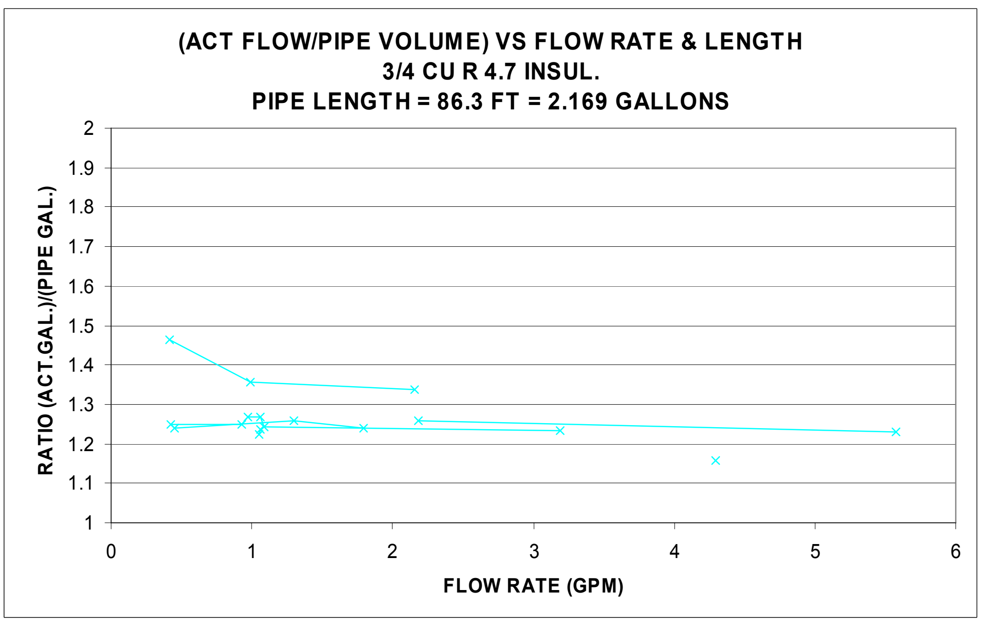


Figure 5-15
Measured AF/PV Ratio vs Flow Rate and TDR For ¾ Inch Rigid Copper Pipe With R-4.7 Insulation– 68.3 Ft Length (Upper Plot TDR = 0.178 – 0.199, Lower Plot TDR = 0.450 – 0.575)

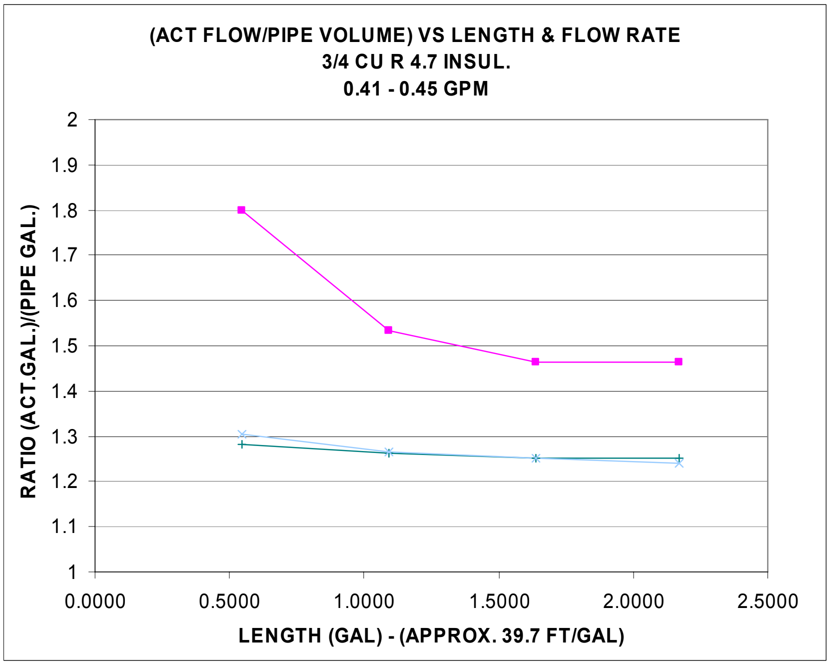


Figure 5-16
Measured AF/PV Ratio vs Pipe Length and TDR For ¾ Inch Rigid Copper Pipe With R-4.7 Insulation– 0.41 – 0.45 GPM (Upper Plot TDR = 0.178, Lower Plot TDR = 0.41 – 0.47)

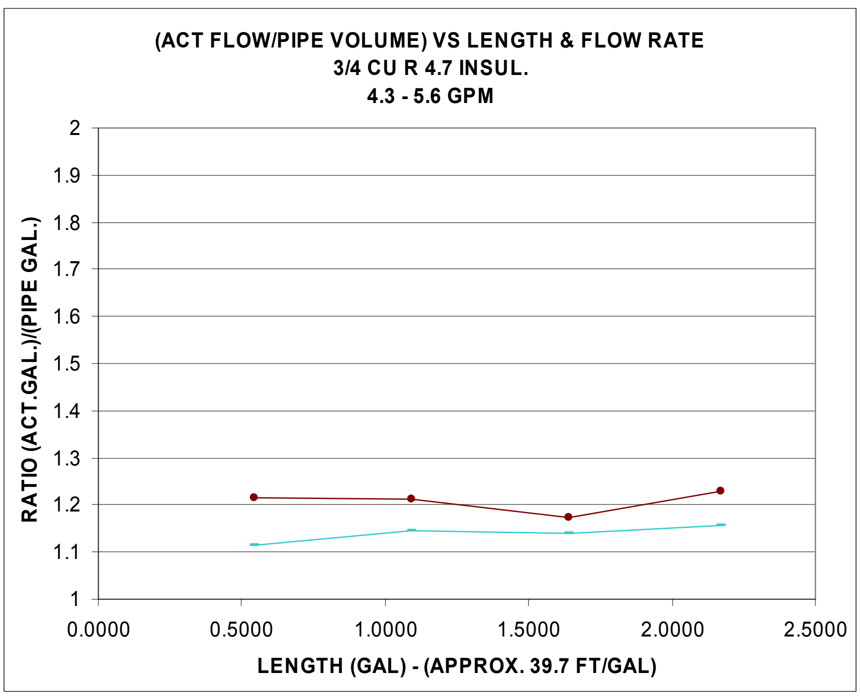


Figure 5-17
Measured AF/PV Ratio vs Pipe Length and TDR For ¾ Inch Rigid Copper Pipe With R-4.7 Insulation– 4.3 – 5.6 GPM (Upper Plot TDR = 0.399, Lower Plot TDR = 0.575)

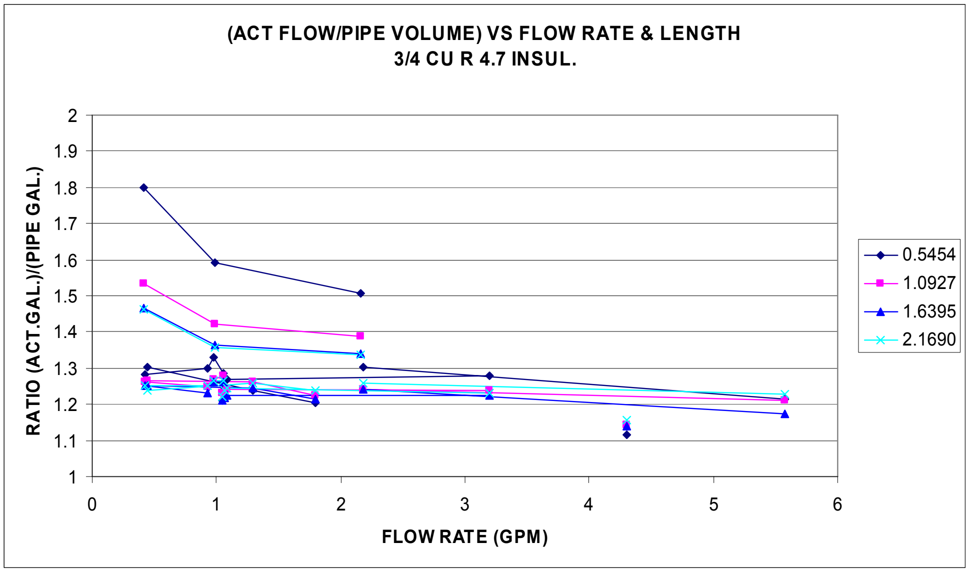


Figure 5-18
Measured AF/PV Ratio vs Flow Rate, Length & TDR For ¾ Inch Rigid Cu Pipe With R-4.7 Insulation– All Lengths and TDRs (Legend is length in gallons, where 39.78 ft = 1 gallon)

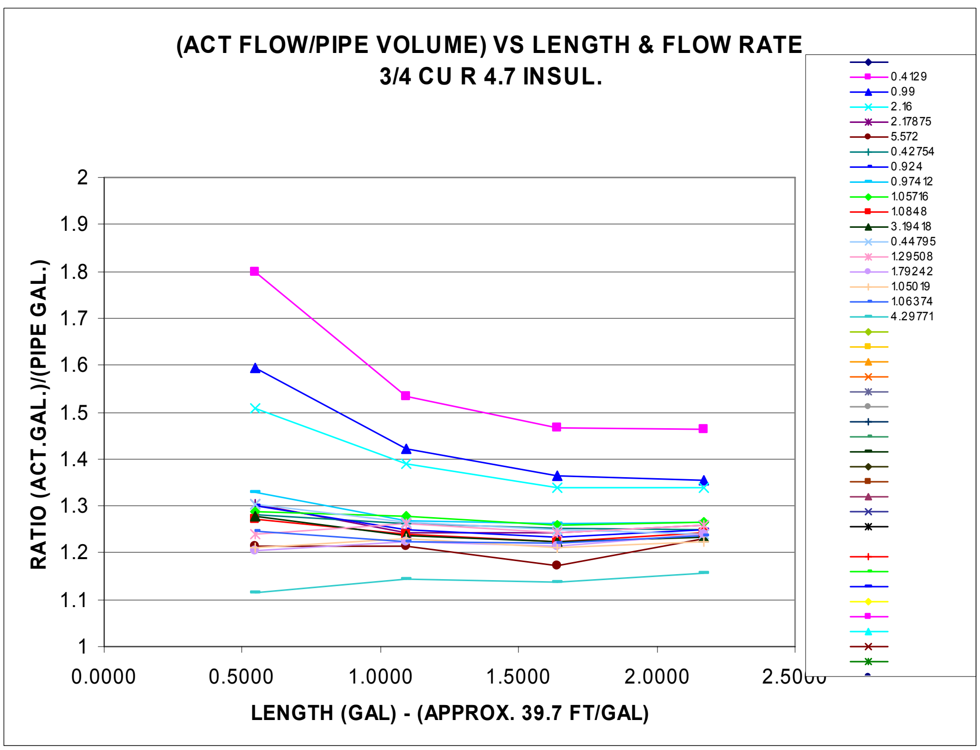


Figure 5-19
Measured AF/PV Ratio vs Pipe Length, Flow Rate and TDR For ¾ Inch Rigid Copper Pipe With R-4.7 Insulation– All Flow Rates and TDRs (Legend is flow rate in GPM)

Comparing AF/PV Results for Bare, R-2.9, and R-4.7 Insulation on ¾ Inch Rigid Copper Pipe

While there was a large and readily apparent difference in the pipe heat loss rate (UA value) on insulated vs uninsulated pipe, the impact of insulation on AF/PV ratios and hence water and energy waste) during the hot water delivery phase was less pronounced. For the most part, AF/PV ratios were only slightly different for insulated vs uninsulated ¾ inch rigid copper pipe. There were, however, some important distinctions.

One important difference was that for the insulated configurations, water exit temperatures at low flow rates remained well above 105 F, indicating that AF/PV ratios would not go to infinity under normally occurring design and operation conditions, unlike for bare pipe. That is to say, the region A behavior shown in figures 4-2 and 4-3, where heat transfer to ambient significantly affected AF/PV ratio, did not exist for the insulated pipe cases. Heat loss rates were so high in the bare pipe configuration, even for pipes in still air, that for low flow rates it would sometimes be necessary to increase tank setpoint temperature in order to obtain "hot enough (105 F) water from pipes of lengths that would often be encountered in practice. Insulated pipes lost much less heat. In fact, one can use the measured pipe UA values to calculate the critical pipe length beyond which 105 F water cannot be obtained at any selected flow rate. This is done in section 8 comparing the UA values from all test configurations.

Another significant difference in AF/PV ratio occurred because it appeared that adding pipe insulation increases the likelihood that flow will go into the slip-flow regime. This appears to happen because the pipe insulation keeps the inner pipe wall warmer, reducing water viscosity at the wall, thus reducing shear resistance and enabling slip-flow to occur more readily than for bare pipe under similar initial temperature conditions. As noted in section 4, slip-flow reduces temperature degradation in the hot water traveling to the fixtures, thus reducing water and energy waste during the delivery phase of hot water flow.

6
TEST RESULTS – HORIZONTAL 1/2 INCH RIGID COPPER PIPE IN AIR

Test and calculational procedures and the flow and temperature test matrix for the horizontal in-air tests are described in Appendix B and will not be repeated here. Detailed presentation of test data, complete with photographs of the test setups and results summary tables for all pipe lengths, flow rates, and TDR cases tested are given in Appendix D. Only summary results are presented here.

The 1/2 inch rigid copper piping tests were done on a test section having six straight lengths each slightly over 20 feet long, plus five U-bends. Immersion thermocouples were inserted in the center of each U-bend, as shown in figures B-3, D-1 and D-2. Total test section length was 128.2 feet.

Piping Heat Loss UA Values

Appendix D shows detailed results for UA values determined during testing on 1/2 inch rigid copper pipe in air with no insulation, ½ inch thick foam insulation (R-3.1 hr ft^2 F/Btu), and ¾ inch thick foam insulation (R-5.2). Those results show that UA is a function of both the temperature difference between the pipe and the surrounding air and the flow rate. In general UA values are higher at higher temperature differences, meaning that pipe heat loss is greater at higher temperature differences in a non-linear manner. For practical engineering calculations, however, we can curve-fit the highest UA values observed (generally the higher temperature difference cases), and use those values for most calculations as if they were independent of temperature difference. Figure 6-1 show scatter-plots of all the data points for the various UA values vs flow rate for all the tests, plus curve-fit lines through the highest UA values vs flow rate for each insulation level. Note that variation of UA with temperature difference is most pronounced for bare pipe, and is much smaller for insulated pipes. This is probably because the actual pipe surface temperature for heat transfer varies more with flow rate in the bare pipe case than do the insulation external surface temperatures. Note especially that the $UA_{zero\text{-}flow}$ for bare pipe varies substantially, and this value should theoretically not be a function of temperature difference between the water in the pipe and the air, because each cool-down test sees continuously dropping pipe temperature, and hence continuously dropping temperature difference. One therefore has substantial latitude in what value one uses to curve-fit the $UA_{zero\text{-}flow}$ data point for the bare pipe case.

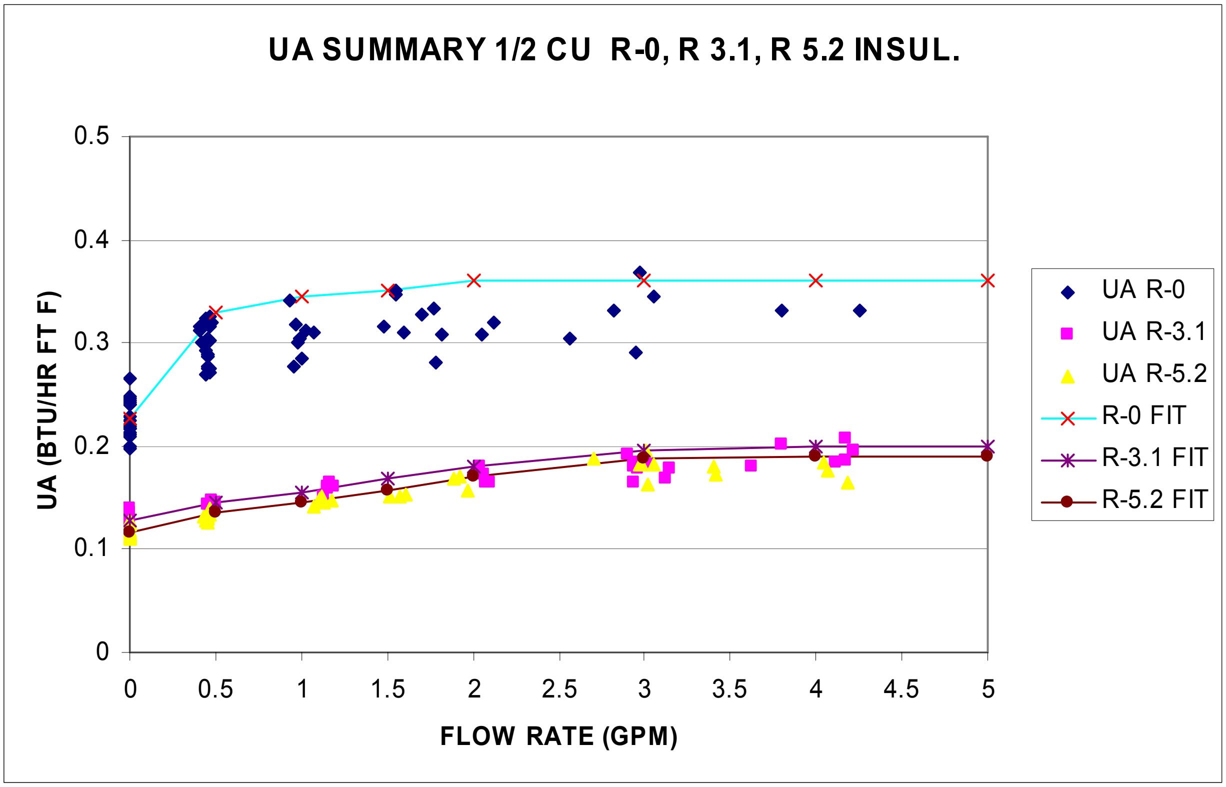


Figure 6-1
Measured Pipe Heat Loss UA Values for 1/2 inch Rigid Copper Pipe

Figure 6-1 shows that for 1/2 in rigid copper pipe, UA values are a function of water flow rate, but they level out and become constant at flow rates above about 3.0 gpm. It also shows that UA values at zero flow are generally somewhat lower than for cases where there is flow

Similar to what was observed on ¾ inch rigid copper pipe, two important observations regarding the value of insulation can be drawn. The first is that adding even a relatively thin layer of insulation (1/2 inch thick foam) cuts pipe heat loss rate almost in half. The second is that using thicker insulation provides only slightly better performance. That is to say, the first small amount of pipe insulation provides a large reduction in pipe heat loss, and increasing insulation thickness provides highly non-linearly increasing benefits, with percentage improvement decreasing rapidly with increasing insulation levels. This is the same behavior that was seen with insulation on ¾ inch rigid copper pipe.

Delivery-Phase AF/PV Ratio

Section 4 contains a discussion of the general types of behaviors of AF/PV ratio that were seen during testing, so that discussion will not be repeated here. Appendix D gives complete data summary tables for all valid tests. This section focuses on presenting quantitative summary results.

1/2 Inch Bare Rigid Copper Pipe AF/PV Results

Figures 6-2 and 6-3 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the 1/2 inch bare rigid copper pipe tests. Figure 6-2 shows results for a pipe length of 21.1 ft (the end of the first straight section), corresponding to 0.256 gallons pipe volume. Figure 6-3 shows results for a pipe length of 128.2 ft – the end of the entire test section, corresponding to 1.5531 gallons pipe volume. Figures 6-4 and 6-5 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.5 and 4.0 gpm respectively.

The individual plots in figures 6-2 through 6-5 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.189 – 0.20) and lower lines corresponding to the highest TDRs tested (nominally 0.450 – 0.50, with one test at 0.738). The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

Examining figure 6-2, we see that for this pipe configuration, AF/PV ratio was high in the entrance region at low flow rates, and decreased with increasing flow rate (up to a limit – see the second paragraph down), especially at low TDR. This corresponds to region B as discussed in Figures 4-2 and 4-3, where flow stratification and pipe axial heat conduction have a detrimental effect on AF/PV ratio. At higher TDR the higher entrance region AF/PV ratios largely disappeared. Note that at low TDR, AF/PV decreases with pipe length. This means that the detrimental effects of stratification and pipe axial heat conduction that occur in the pipe entrance region are largely negated if the pipe is long enough.

Unlike the ¾ inch rigid copper tests, we also see an increase in entrance-region AF/PV ratios at high flow rates and low TDR. This test configuration was the only one where this phenomenon was observed. It is possible that this behavior is an artifact of the difficulty in obtaining accurate measurements at high flow rates in the entrance regions of small diameter pipes. If a real effect, it may correspond to a flow regime of high turbulence caused by high flow rates, causing high mixing even in the entrance region of the pipe. This corresponds to region C as discussed in Figures 4-2 and 4-3. This effect was not observed at longer pipe lengths, so if real, its effect appears to disappear with length. We will look for this phenomenon to occur in future tests on other pipe configurations to improve our understanding.

Examining figure 6-3 (long pipe length) at low flow rate, and figure 6-4 (low flow rates) at long pipe lengths, we see that at low TDR, AF/PV ratio increases after having passed through a minimum at shorter pipe lengths. This corresponds to region A as discussed in figures 4-2 and 4-3, where heat transfer to ambient is an important effect. It is apparent that if the pipe had been slightly longer, or the surrounding air had been slightly colder, AF/PV ratio would have gone to infinity. This was in fact observed in some of the set-up tests for this pipe configuration while attempting to adjust thermocouple positioning, before rigorous data collection began on the configuration.

It is worthwhile to note that there was a fairly common slight inconsistency in the temperature measurements at the third U-bend temperature measuring location (0.75 gallon location on figure 6-4) compared to the other temperature measuring stations. This reflects the difficulties in properly aligning the thermocouples in the pipe T's, as discussed in Appendix B. It took numerous tries to get the thermocouples to read as consistently as they did. It is probable that the thermocouple at this location was close to the pipe wall, rather than centered in the pipe T in which it was located.

Perhaps the most interesting results were seen in the pipe entrance region at high flow rates and high TDRs. Examining the lower curve in figures 6-2 and 6-5, we see that at high flow rates, the AF/PV ratio actually dropped to 1.0, indicating almost perfect plug-flow in the pipe with little degradation in the hot water temperature as it traveled through the pipe. These were the first tests where the slip-flow phenomenon, described as region D in figures 4-2 and 4-3, was clearly observed. It appears that in these tests the slip-flow occurred in the entrance region, and continued further down the pipe at higher flow rates. It appears from Figure 6-5 that the slip-flow had largely disappeared by the time the flow had reached the end of the test section, but the reduction in AF/PV brought about in the entrance region lowered the net AF/PV ratio at the longer pipe lengths. Note that this slip-flow develops at higher TDRs, which result from the combination of high entering hot water temperatures and high initial pipe temperatures. As seen in figure 6-5, at lower TDRs, the slip-flow does not develop, and the AF/PV ratios behave as had been seen for the majority of the ¾ inch rigid copper pipe tests. This means that pipe internal mixing effects and water-to-pipe-mass heat transfer are important effects at low TDRs, but are largely negated as TDR increases, especially when slip-flow develops. Note also that, as seen in the lowest plot of figure 6-4, which was a case of very high TDR, slip-flow was close to developing even at flow rates as low as 0.5 GPM. The flow rates at which slip-flow can develop are a function of the TDR, as is the pipe length over which the slip-flow will persist. Whether or not slip-flow will develop, and at what flow rates and TDRs, and over what pipe length the slip-flow will persist are also functions of the pipe diameter, smoothness of the inner pipe wall, presence or absence of fittings that can cause turbulence, and the pipe material (affects water adhesion to the wall and hence shear resistance at the water/wall interface).

Figures 6-6 and 6-7 show all AF/PV results from all tests on bare 1/2 inch rigid copper piping, plotted on the same plots. In figure 6-6 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs. Unlike for the ¾ rigid copper, however, once slip-flow begins to develop, the lower curves can be for the shorter pipe lengths, not just longer ones. If slip-flow has not developed, the lower plots normally correspond to longer pipe lengths. In figure 6-7 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix D allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 6-6 and 6-7 that for ½ inch bare rigid copper piping, AF/PV ratios normally range from a low of around 1.1 to a high of around 2.0, with most in the range of 1.2 to 1.6.

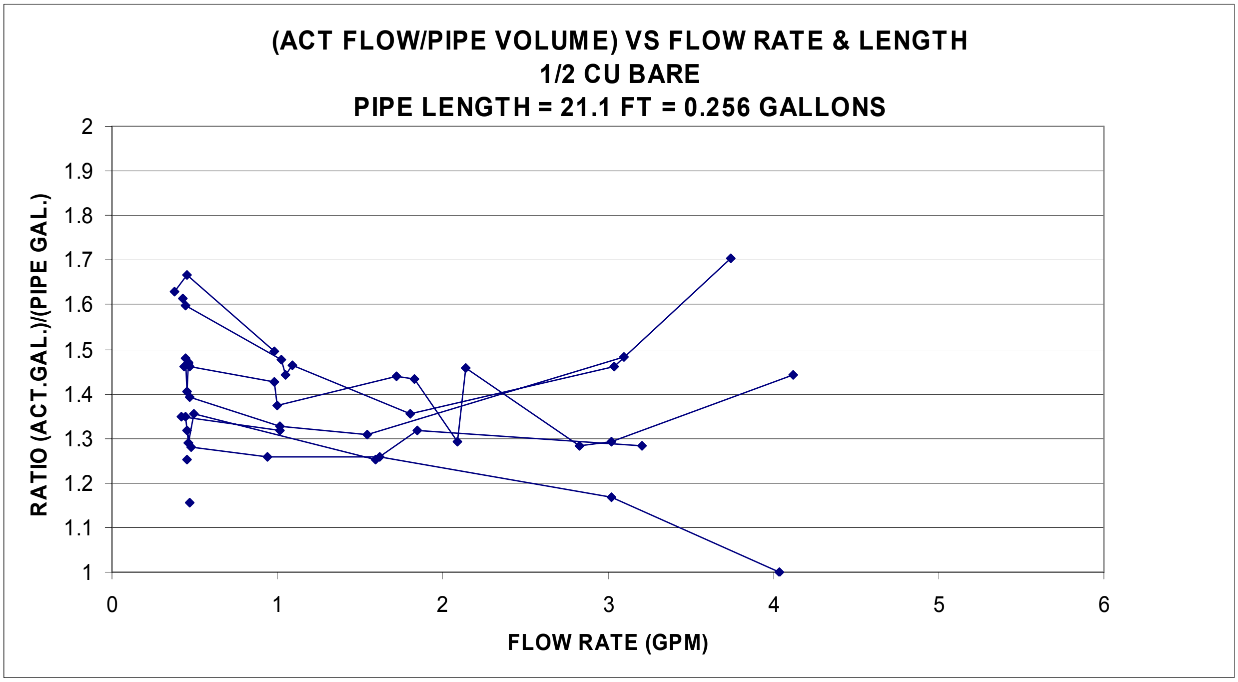


Figure 6-2
Measured AF/PV Ratio vs Flow Rate and TDR For Bare 1/2 Inch Rigid Copper Pipe – 21.1Ft Pipe Length (Upper Plot TDR = 0.189 – 0.196, Lower Plot TDR = 0.461 – 0.473)

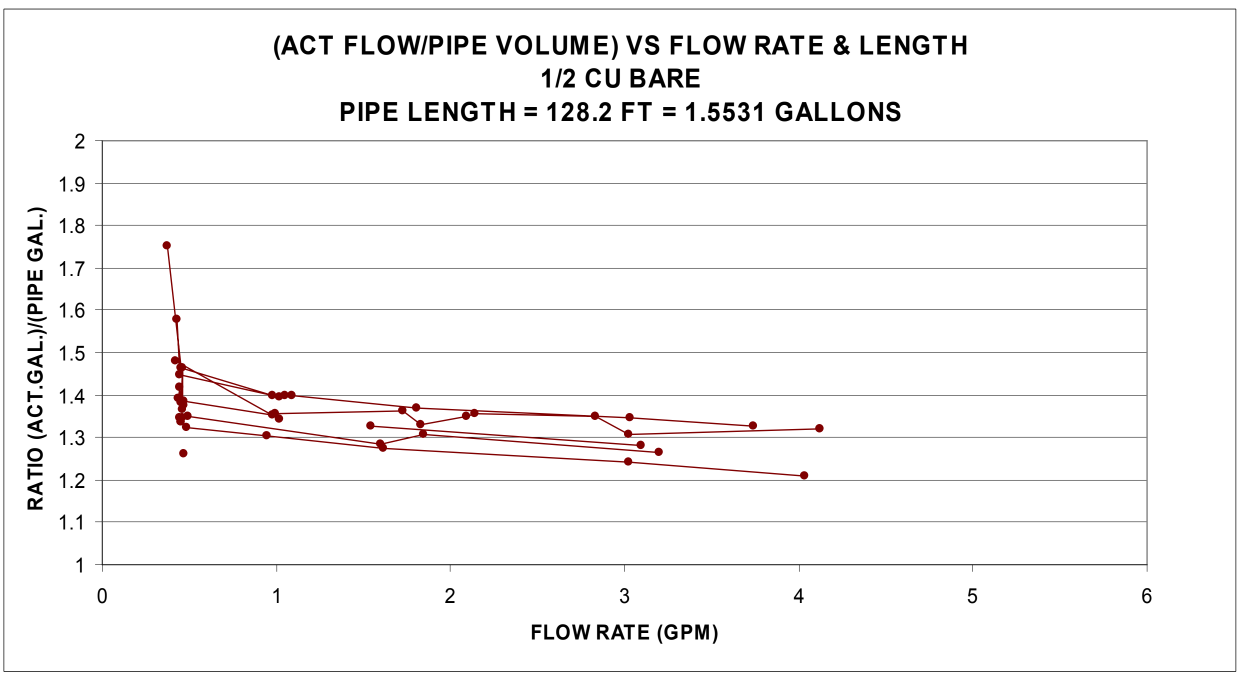


Figure 6-3
Measured AF/PV Ratio vs Flow Rate and TDR For Bare 1/2 Inch Rigid Copper Pipe – 128.2 Ft Pipe Length (Upper Plot TDR = 0.189 – 0.196, Lower Plot TDR = 0.461– 0.473

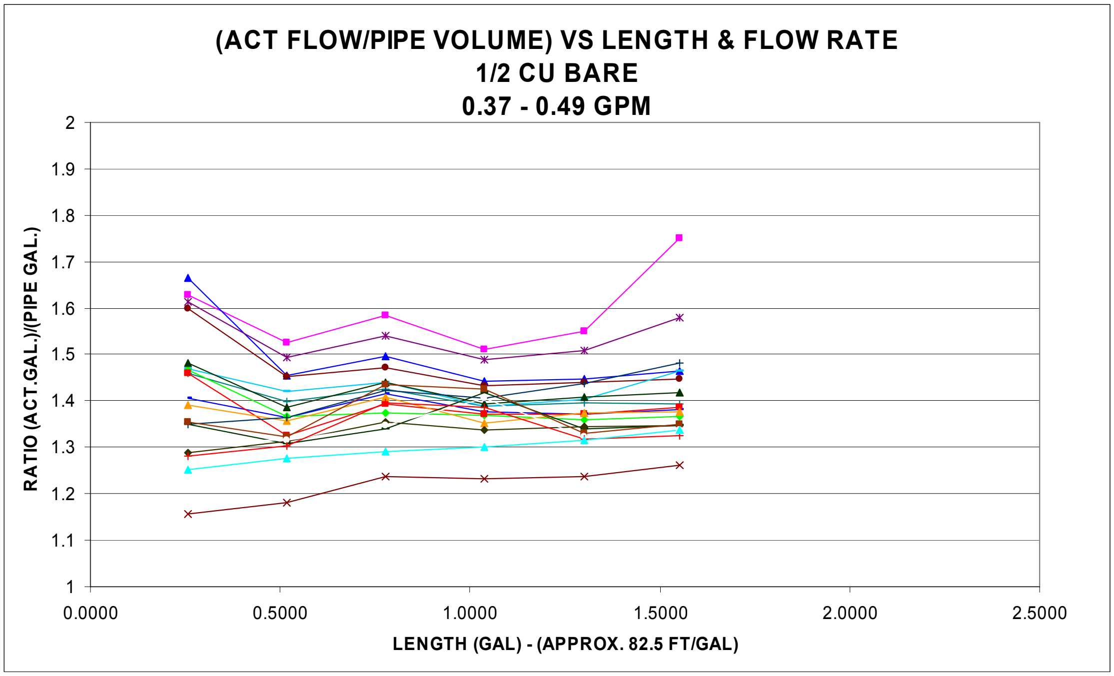


Figure 6-4
Measured AF/PV Ratio vs Pipe Length and TDR For Bare 1/2 Inch Rigid Copper Pipe – 0.37 -0.49 GPM Flow Rate (Upper Plot TDR = 0.189, Lower Plot TDR = 0.738)

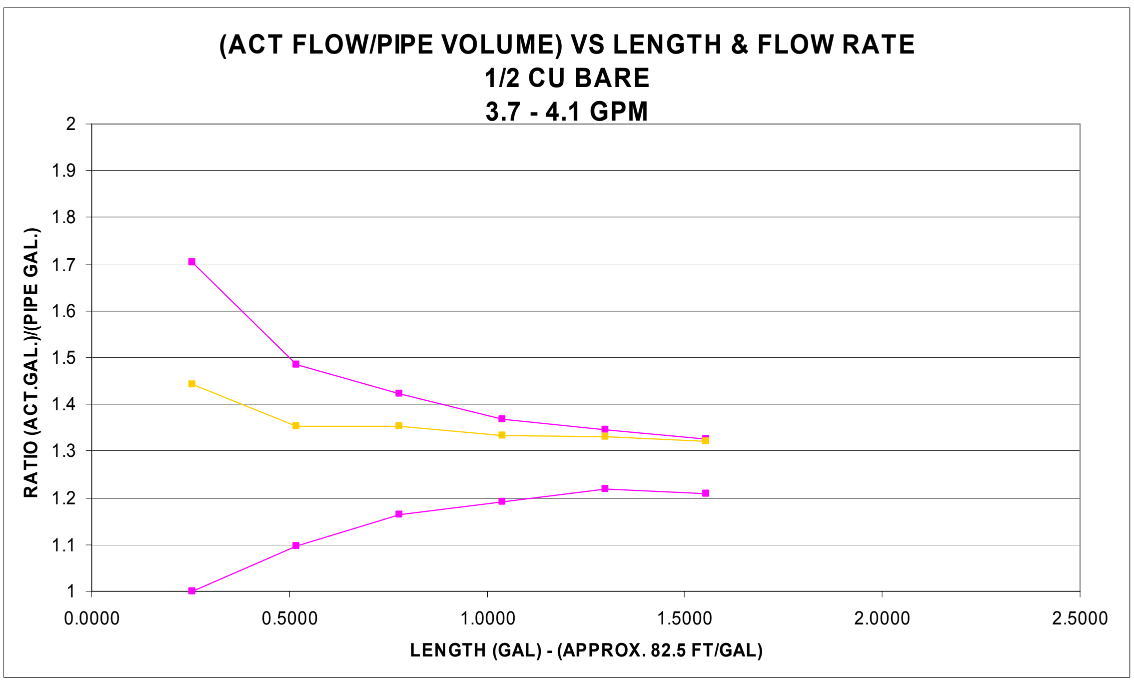


Figure 6-5
Measured AF/PV Ratio vs Pipe Length and TDR For Bare 1/2 Inch Rigid Copper Pipe – 3.7 - 4.1 GPM Flow Rate (Upper Plot TDR = 0.224, Lower Plot TDR = 0.466)

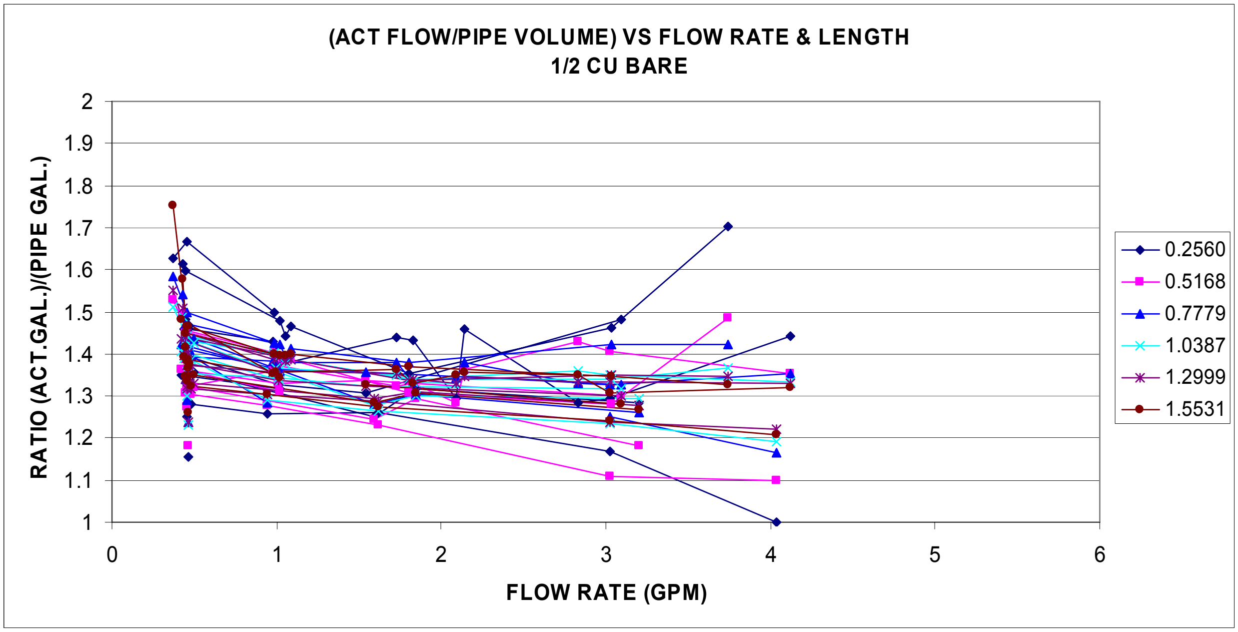


Figure 6-6
Measured AF/PV Ratio vs Flow Rate, Length & TDR For Bare 1/2 Inch Rigid Copper Pipe – All Pipe Lengths and TDRs (Legend is pipe length in gallons, where 82.5 ft = 1 gallon)

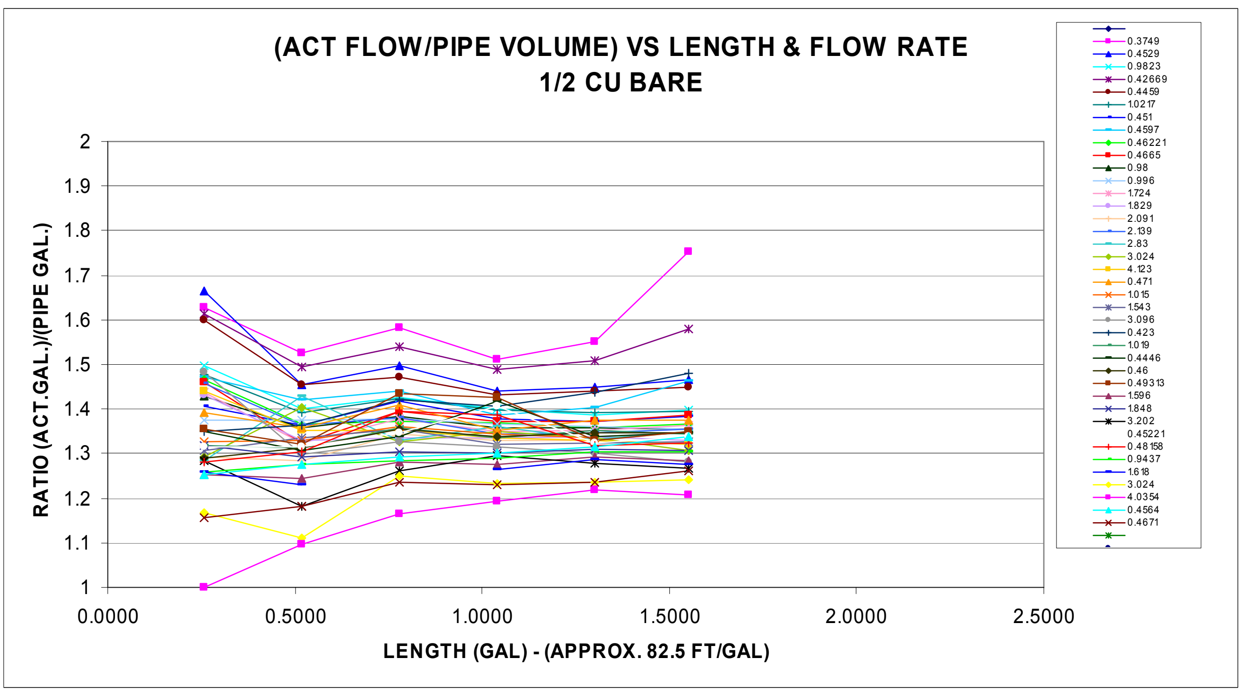


Figure 6-7
Measured AF/PV Ratio vs Pipe Length, Flow Rate and TDR For Bare 1/2 Inch Rigid Copper Pipe – All Flow Rates and TDRs (Legend is flow rate in gpm.)

1/2 Inch Rigid Copper Pipe With ½ Inch Thick Foam Insulation AF/PV Results

The foam pipe insulation used for these tests was of the same type most commonly seen in the field site visits - black closed-cell polyethylene or polyolefin foam. Properties for these two materials are similar. R-values were printed on the insulation, like was observed at the field sites. The typical thermal conductivity listed by various manufacturers for these types of pipe insulation was around k = 0.02 Btu/hr ft F. The R-values ranged as shown in table 3-2 for the different foam thicknesses and different pipe sizes, because R-value is based on the outer diameter of the foam. The above listed thermal conductivity can be used in standard heat transfer calculations for multi-layer cylindrical objects (given in most introductory heat transfer textbooks) to calculate the R-value for insulation of a given thickness applied to pipe of a given diameter. The R-value of the ½ inch thick foam on the 1/2 inch nominal copper pipe size (approximately 5/8 inch OD) was R-3.1.hr ft^2F/Btu

Figures 6-8 and 6-9 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the tests on 1/2 inch rigid copper pipe with R-3.1 insulation. Figure 6-8 shows results for a pipe length of 21.1 ft (end of first straight section), corresponding to 0.256 gallons pipe volume. Figure 6-9 shows results for a pipe length of 128.2 ft – the end of the entire test section, corresponding to 1.5531 gallons pipe volume. Figures 6-10 and 6-11 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.5 and 4.0 gpm respectively.

The individual plots in figures 6-8 through 6-11 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.185 – 0.2) and lower lines corresponding to the highest TDRs tested (nominally 0.629 – 0.65). The lines proceed generally from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

We see when examining Figure 6-8 that adding pipe insulation has increased the difficulty of obtaining consistently accurate data at shorter pipe lengths. The fairly wide variations of AF/PV ratio with changing flow rate reflect the difficulty in obtaining accurate measurements at high flow rates in short lengths of small diameter pipe, as was discussed in Appendix B. It probably also reflects an increasing tendency for the flow in the entrance region of the pipe to go in and out of slip-flow because the addition of pipe insulation has made slip-flow more likely to occur. Slip-flow is an inherently non-steady-state phenomenon, and flow appears to go in and out of slip-flow under some conditions. As TDR increases, slip-flow becomes more likely to occur. Comparing figures 6-2 and 6-8 we see that the addition of insulation is allowing slip-flow to occur at lower TDRs compared to bare pipe. This is probably because the insulation keeps the inner pipe surface warmer, lowering viscosity near the wall thus reducing shear resistance.

Comparing figures 6-8 and 6-9, we see that data is better-behaved at longer pipe lengths.

The 1/2 rigid copper with R-3.1 insulation tests showed results similar to the uninsulated tests with several exceptions. First, at lower flow rates, AF/PV ratios were slightly lower for the R-3.1 tests compared to the bare tests, at similar TDRs, but the difference was not as great as was observed for the ¾ inch rigid copper tests. This suggests that heat transfer to ambient was affecting the AF/PV ratio of the low-flow-rate bare-pipe tests even at short pipe lengths, but not as much as in the bare ¾ inch copper. Comparing figures 6-3 and 6-4 to figures 6-9 and 6-10, we see that adding insulation significantly reduced the impact of heat transfer to ambient on the AF/PV ratio at longer pipe lengths. (See region A of figures 4-2 and 4-3.)

As in the bare ½ inch rigid copper tests, slip-flow was observed to occur at higher flow rates and high TDRs, especially in the entrance region of the test section. As seen in the lower plots on figures 6-8 through 6-13, at high flow rates and short pipe lengths, AF/PV ratio actually went to 1.0 in several tests. This was identified as region D in figures 4-2 and 4-3. It appears that the slip-flow was able to persist over longer pipe lengths in the insulated pipe compared to uninsulated pipe. However, flow still degraded to normally developing flow before the end of the entire test section. There was, however, a net benefit of reduced AF/PV for the longer lengths from the AF/PV ratios having been so low in the entrance regions due to slip-flow.

Figures 6-12 and 6-13 show all AF/PV results from all tests on 1/2 inch rigid copper piping with R-3.1 insulation, plotted on the same plots. In figure 6-12 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs. Unlike for the ¾ rigid copper, however, once slip-flow begins to develop, the lower curves can be for the shorter pipe lengths. If slip-flow has not developed, the lower plots normally correspond to longer pipe lengths. In figure 6-13 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix D allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 6-12 and 6-13 that for 1/2 inch rigid copper piping with R-3.1 insulation AF/PV ratios mostly behaved similarly to the bare pipe case, with the normal AF/PV ratios observed being in the range of 1.1 to 2.0, and most being in the range of 1.2 to 1.6, except when slip-flow occurred.

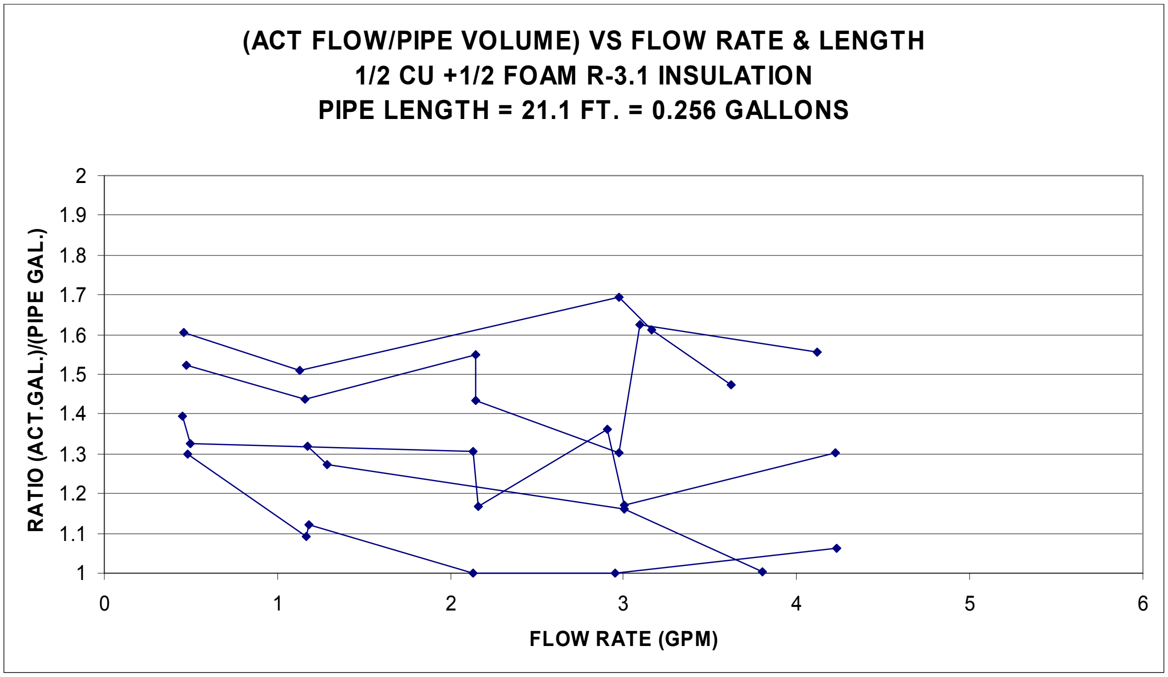


Figure 6-8
Measured AF/PV Ratio vs Flow Rate and TDR For 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation– 21.1 Ft Length (Upper Plot TDR = 0.185 – 0.199, Lower Plot TDR = 0.629 – 0.650)

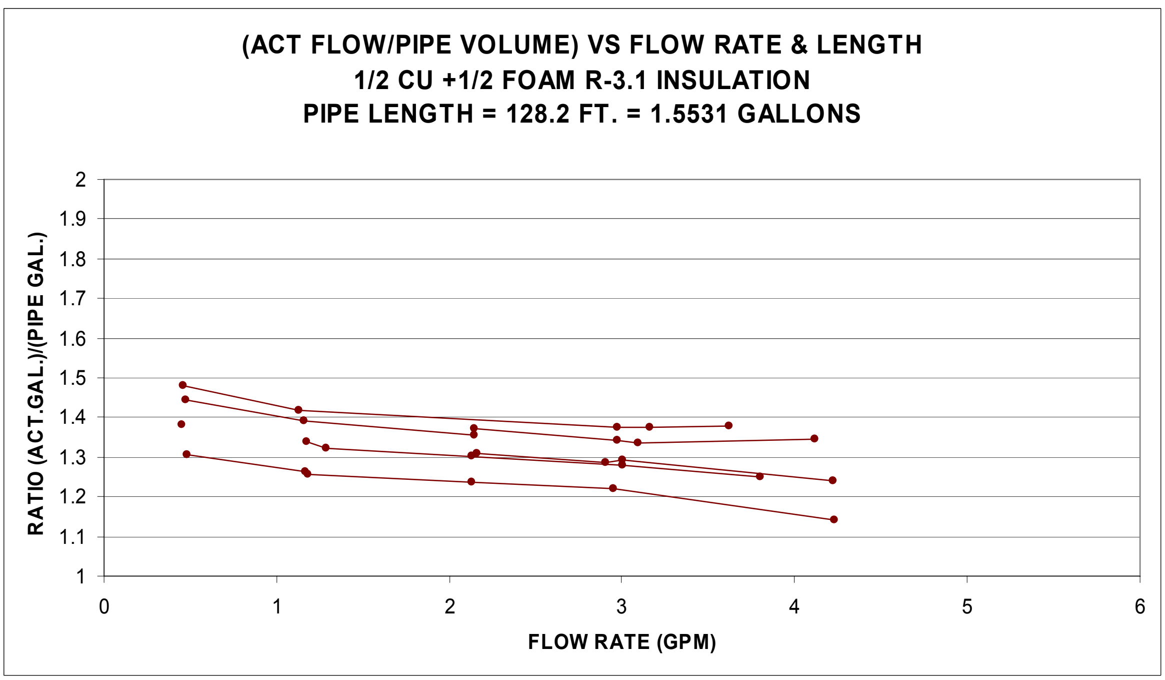


Figure 6-9
Measured AF/PV Ratio vs Flow Rate & TDR For 1/2 Inch Rigid Cu Pipe With R-3.1 Insul. – 128.2 Ft Length (Upper Plot TDR = 0.185 – 0.199, Lower Plot TDR = 0.629 – 0.650)

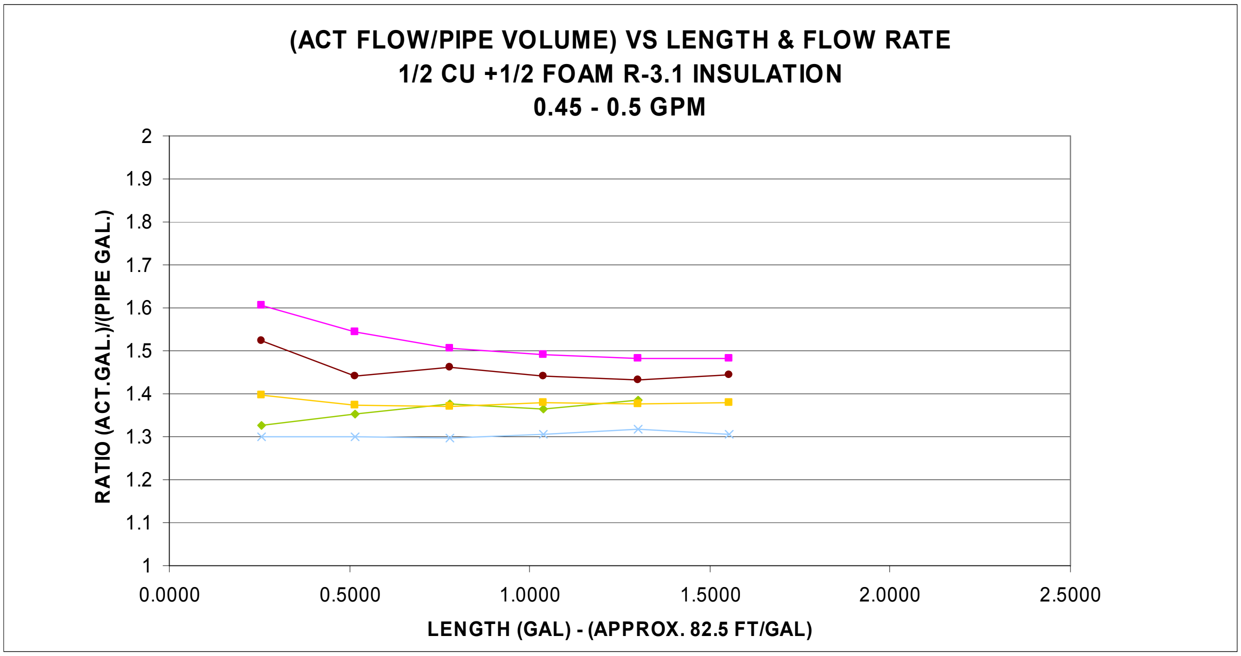


Figure 6-10
Measured AF/PV Ratio vs Pipe Length & TDR For 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation– 0.45 – 0.5 GPM (Upper Plot TDR = 0.185, Lower Plot TDR = 0.629)

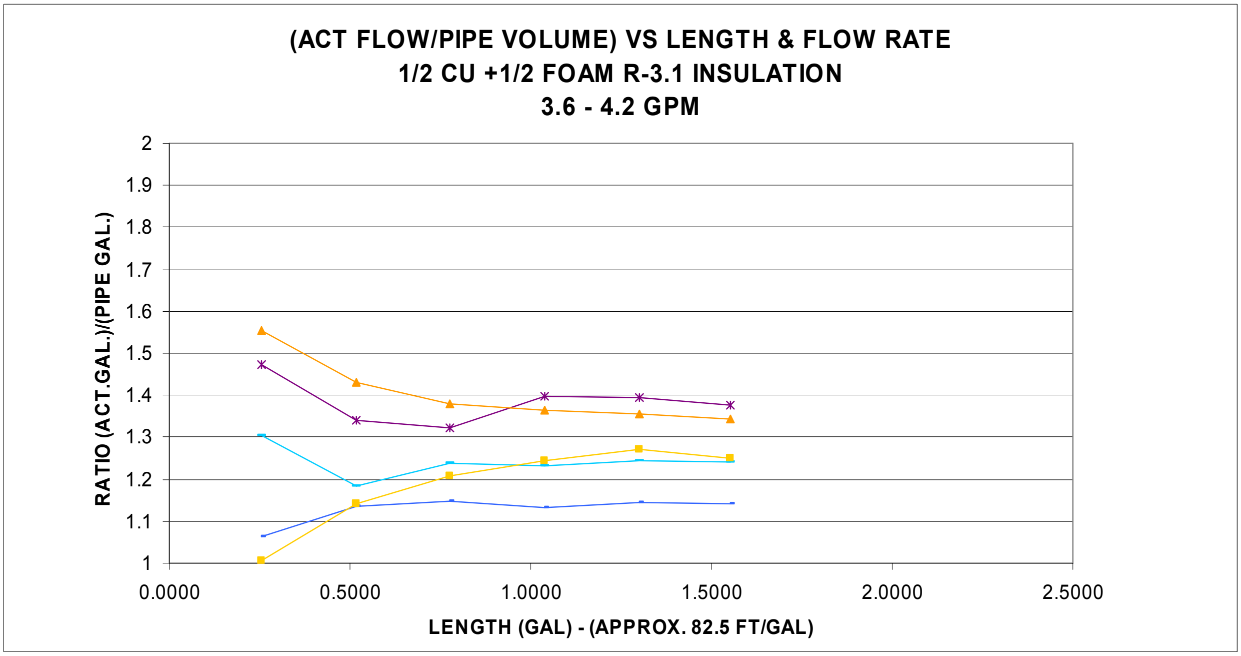


Figure 6-11
Measured AF/PV Ratio vs Pipe Length and TDR For 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation– 3.6 – 4.2 GPM (Upper Plot TDR = 0.199, Lower Plot TDR = 0.650)

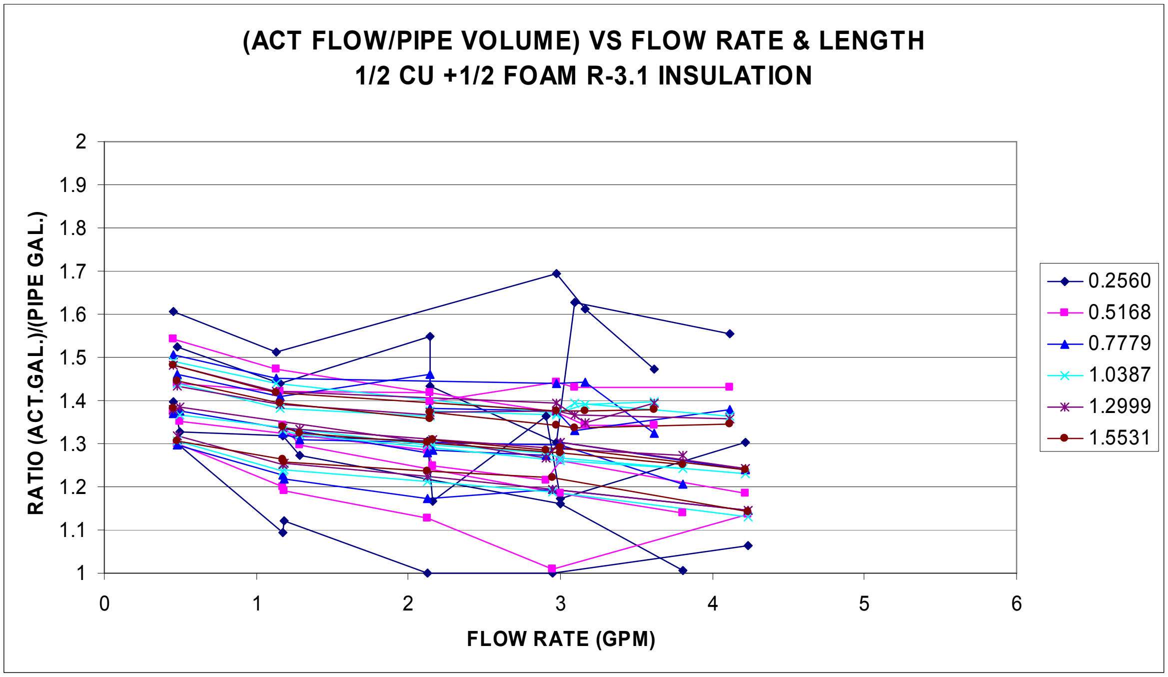


Figure 6-12
Measured AF/PV Ratio vs Flow Rate, Length & TDR For 1/2 Inch Rigid Cu Pipe With R-3.1 Insulation– All Lengths and TDRs (Legend is length in gallons, where 82.5 ft = 1 gallon)

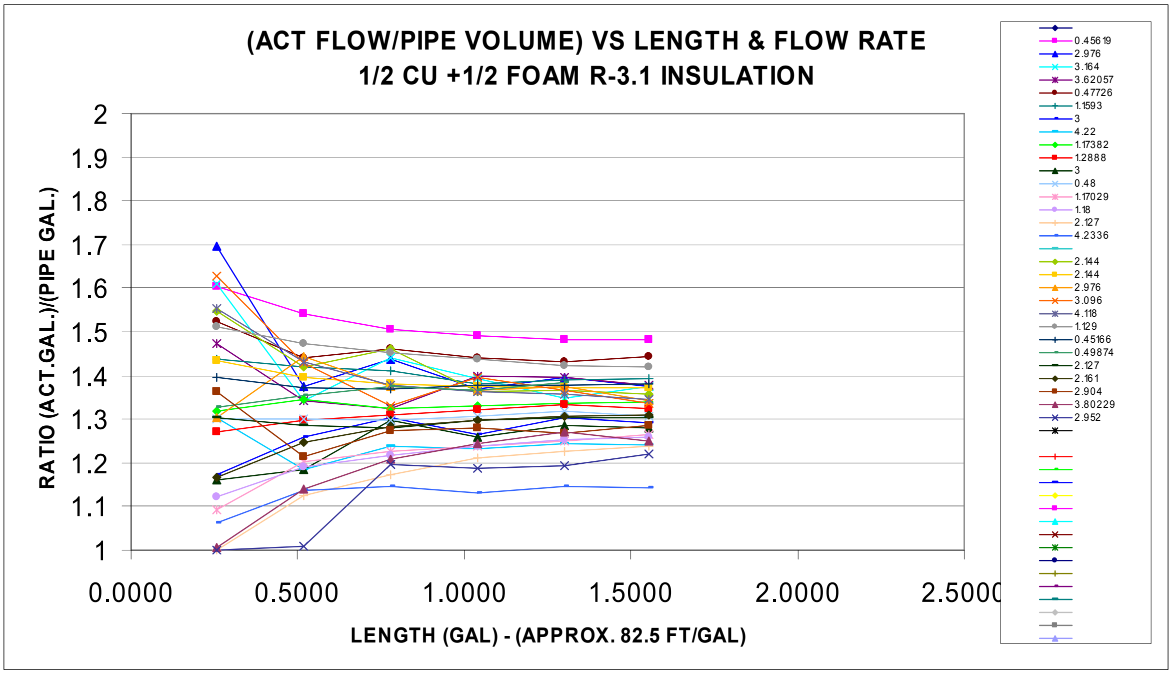


Figure 6-13
Measured AF/PV Ratio vs Pipe Length, Flow Rate, and TDR For 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation– All Flow Rates and TDRs (Legend is flow rate in GPM)

1/2 Inch Rigid Copper Pipe With 3/4 Inch Thick Foam Insulation AF/PV Results

The R-value of the 3/4 inch thick foam on the 1/2 inch nominal copper pipe size (approximately 5/8 inch OD) was R-5.2 hr ft^2F/Btu

Figures 6-14 and 6-15 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the tests on 1/2 inch rigid copper pipe with R-5.2 insulation. Figure 6-14 shows results for a pipe length of 21.1 ft (end of first straight section), corresponding to 0.256 gallons pipe volume. Figure 6-15 shows results for a pipe length of 128.2 ft (end of entire test section), corresponding to 1.5531 gallons pipe volume. Figures 6-16 and 6-17 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.45 and 4.0 gpm respectively.

The individual plots in figures 6-14 through 6-17 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.177) and lower lines corresponding to the highest TDRs tested (nominally 0.7). The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

The 1/2 inch rigid copper with R-5.2 insulation tests showed results similar to the R-3.1 insulated tests discussed above. Differences were that with the thicker R-5.2 insulation, slip-flow began to occur at slightly lower TDRs, and lower flow rates, and persisted over longer pipe lengths than the R-3.1 case.

Figures 6-18 and 6-19 show all AF/PV results from all tests on 1/2 inch rigid copper piping with R-5.2 insulation, plotted on the same plots. In figure 6-18 upper lines correspond to lower TDRs and shorter pipe lengths, while lower lines correspond to higher TDRs. Unlike for the ¾ rigid copper, however, once slip-flow begins to develop, the lower curves can be for the shorter pipe lengths. If slip-flow has not developed, the lower plots normally correspond to longer pipe lengths. In figure 6-19 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. The complete summary data tables included in Appendix D allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 6-18 and 6-19 that for 1/2 inch rigid copper piping with R-5.2 insulation AF/PV ratios mostly behaved similarly to the bare pipe case, with the normal AF/PV ratios observed being between 1.1 and 2.0, and mostly between 1.2 and 1.5, except when slip-flow occurred.

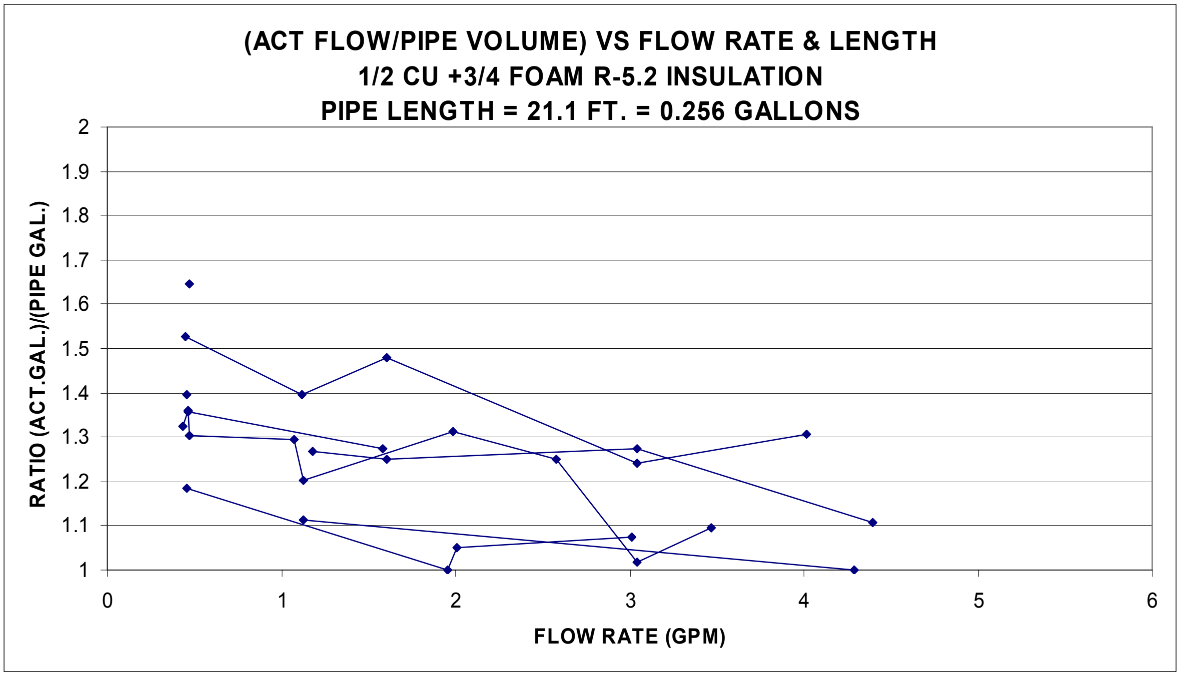


Figure 6-14
Measured AF/PV Ratio vs Flow Rate & TDR For 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation– 21.1 Ft Length (Upper Plot TDR = 0.178 – 0.199, Lower Plot TDR = 0.450 – 0.575)

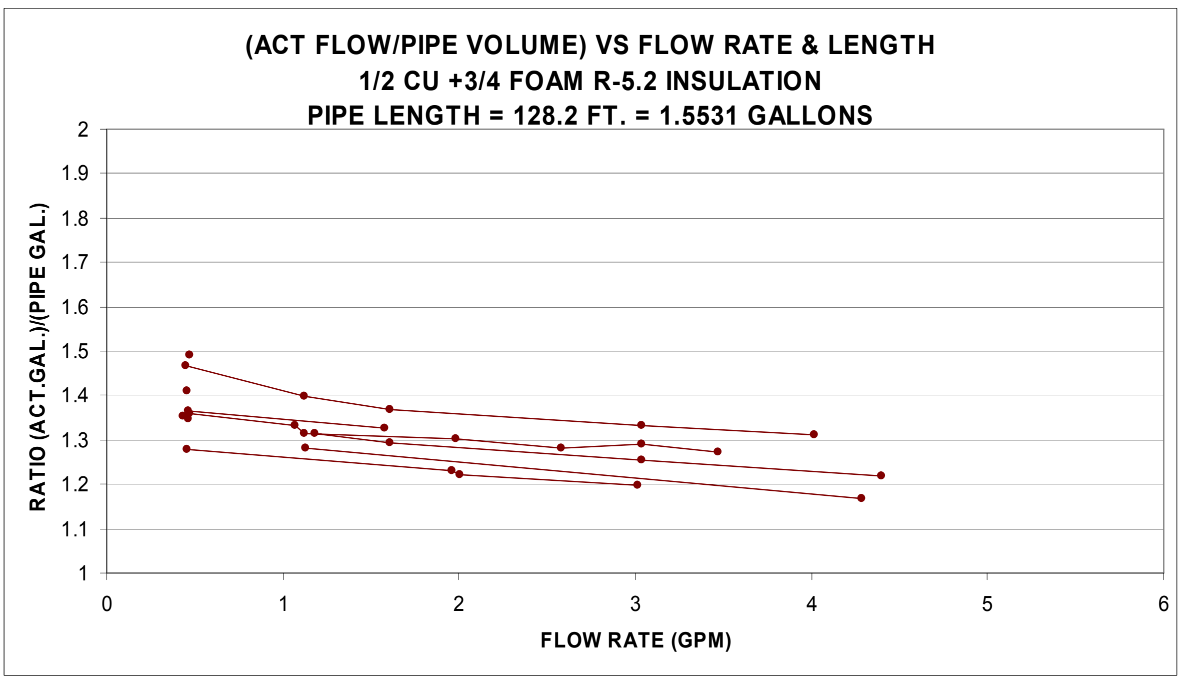


Figure 6-15
Measured AF/PV Ratio vs Flow Rate & TDR For 1/2 Inch Rigid Copper Pipe With R-5.2 Insul. – 128.2 Ft Length (Upper Plot TDR = 0.178 – 0.199, Lower Plot TDR = 0.450 – 0.575)

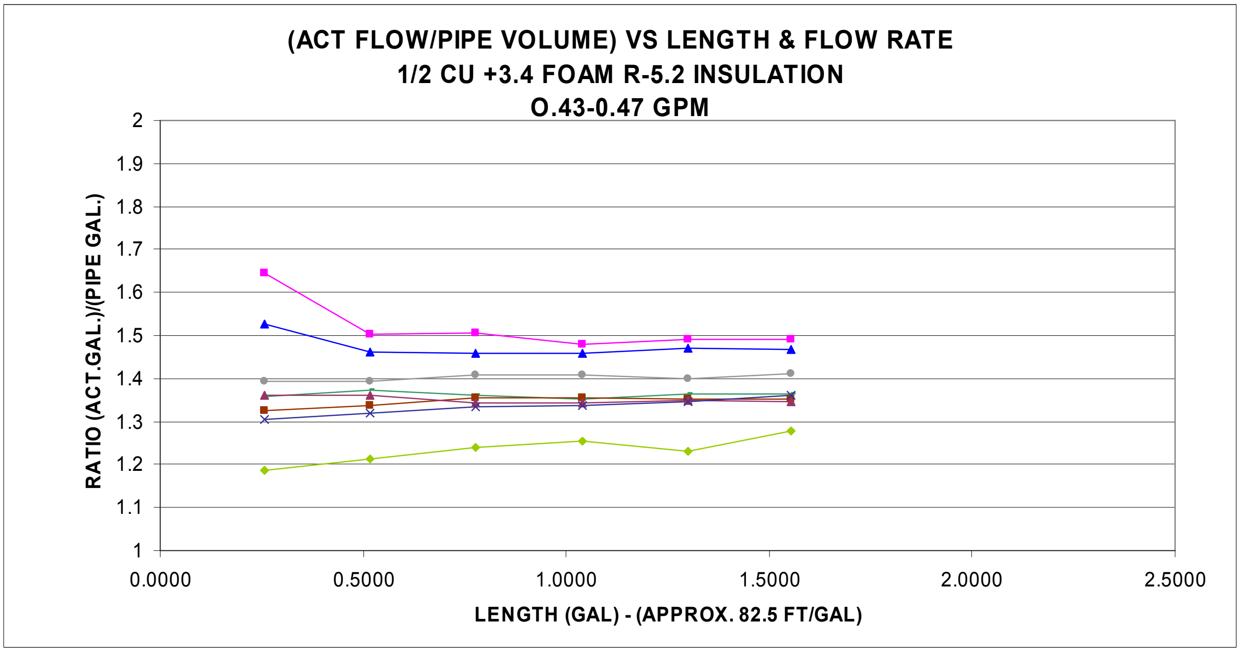


Figure 6-16
Measured AF/PV Ratio vs Pipe Length & TDR For 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation– 0.43 – 0.47 GPM (Upper Plot TDR = 0.178, Lower Plot TDR = 0.41 – 0.47)

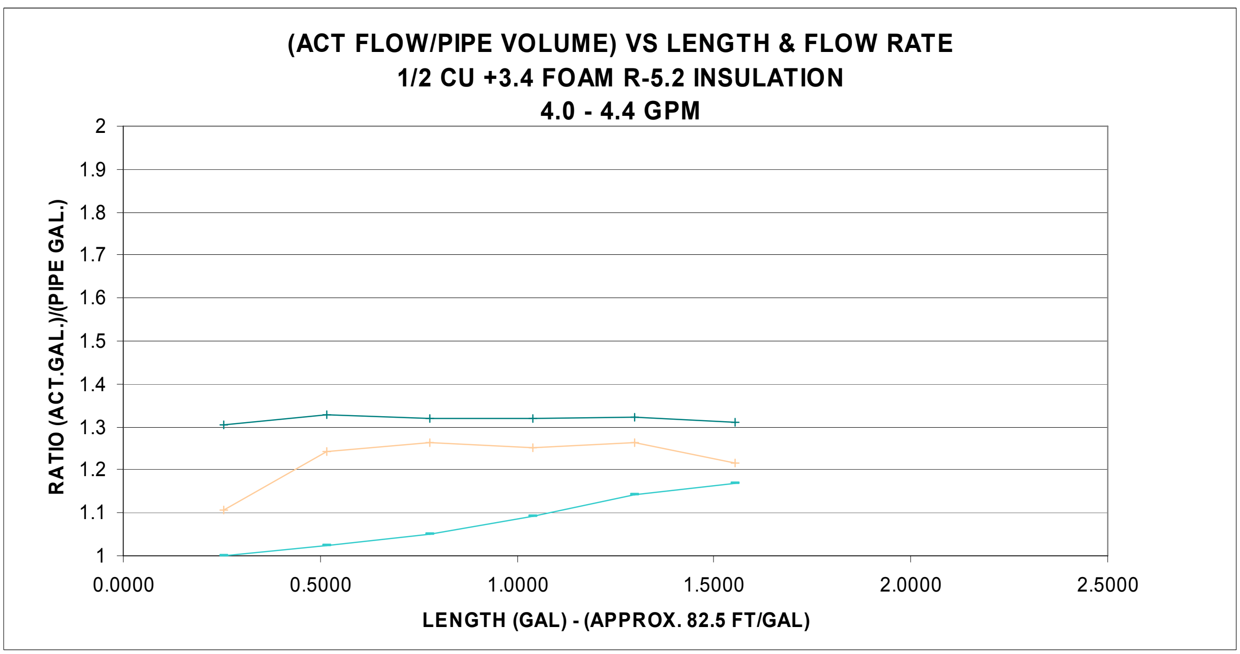


Figure 6-17
Measured AF/PV Ratio vs Pipe Length & TDR For 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation– 4.0 – 4.4 GPM (Upper Plot TDR = 0.399, Lower Plot TDR = 0.575)

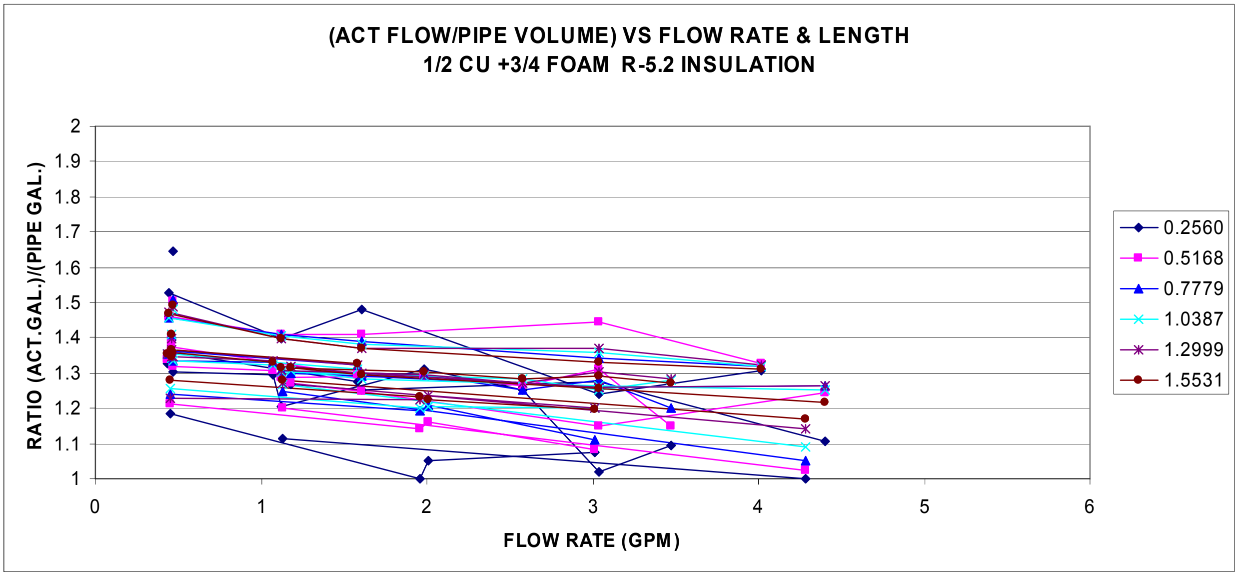


Figure 6-18
Measured AF/PV Ratio vs Flow Rate, Length & TDR For 1/2 Inch Rigid Cu Pipe With R-5.2 Insulation– All Lengths and TDRs (Legend is length in gallons where 82.5 ft = 1 gallon)

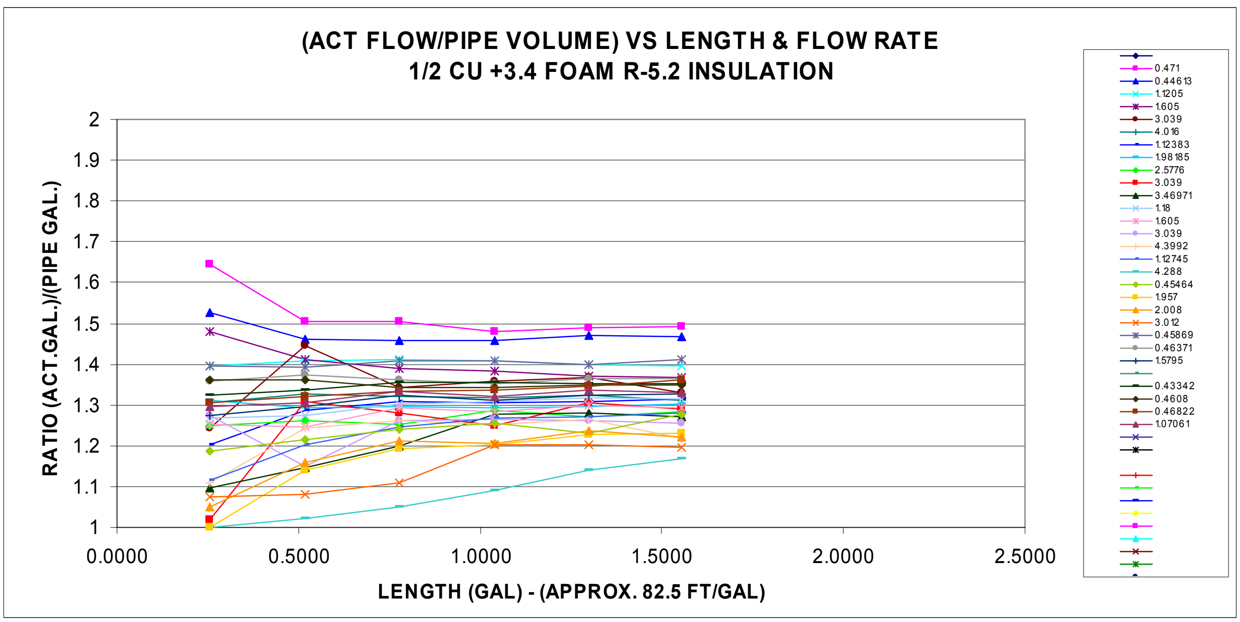


Figure 6-19
Measured AF/PV Ratio vs Pipe Length, Flow Rate, and TDR For 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation– All Flow Rates and TDRs (Legend is flow rate in GPM)

Comparing AF/PV Results for Bare, R-3.1, and R-5.2 Insulation on 1/2 Inch Rigid Copper Pipe

Behavior trends for the ½ inch rigid copper piping were similar to those for the ¾ inch rigid copper piping.

While there was a large and readily apparent difference in the pipe heat loss rate (UA value) on insulated vs uninsulated pipe, the impact of insulation on AF/PV ratios and hence water and energy waste) during the hot water delivery phase was less pronounced. For the most part, AF/PV ratios were only slightly different for insulated vs uninsulated ½ inch rigid copper pipe. There were, however, some important distinctions.

One important difference was that for the insulated configurations, water exit temperatures at low flow rates remained well above 105 F, indicating that AF/PV ratios would not go to infinity under normally occurring design and operation conditions, unlike for bare pipe. That is to say, the region A behavior shown in figures 4-2 and 4-3, where heat transfer to ambient significantly affected AF/PV ratio, did not exist for the insulated pipe cases. Heat loss rates were so high in the bare pipe configuration, even for pipes in still air, that for low flow rates it would sometimes be necessary to increase tank setpoint temperature in order to obtain "hot enough (105 F) water from pipes of lengths that would often be encountered in practice. Insulated pipes lost much less heat. In fact, one can use the measured pipe UA values to calculate the critical pipe length beyond which 105 F water cannot be obtained at any selected flow rate. This is done in section 8 of this report, comparing the UA values from all test configurations.

Another significant difference in AF/PV ratio occurred because it appeared that adding pipe insulation increases the likelihood that flow will go into the slip-flow regime, and that the slip-flow regime begins at lower TDRs and lower flow rates, and continues for longer pipe lengths as pipe insulation levels are increased. This appears to happen because the pipe insulation keeps the inner pipe wall warmer, reducing water viscosity at the wall, thus reducing shear resistance and enabling slip-flow to occur more readily than for bare pipe under similar initial temperature conditions. As noted in section 4, slip-flow reduces temperature degradation in the hot water traveling to the fixtures, thus reducing water and energy waste during the delivery phase of hot water flow.

7

TEST RESULTS – HORIZONTAL ¾ INCH PAX PIPE IN AIR

Test and calculational procedures and the flow and temperature test matrix for the horizontal in-air tests are described in Appendix B and will not be repeated here. Detailed presentation of test data, complete with photographs of the test setups and results summary tables for all pipe lengths, flow rates, and TDR cases tested are given in Appendix E. Only summary results are presented here.

The 3/4 PAX (PEX-AL-PEX) piping tests were done on a test section having four approximately straight lengths slightly over 22 feet in length, plus three U-bends. Unlike the copper tests, the PAX piping had no fittings at all in the test section. The pipe was merely hand-bent into the U-bends. Immersion thermocouples were inserted in the center of each U-bend, and at the inlet and outlet of the pipe, directly through the pipe sidewall using a custom-fabricated thermocouple clamp/seal as shown in figures B-4 and E-1. Total test section length was approximately 100 feet.

Piping Heat Loss UA Values

Appendix E shows detailed results for UA values determined during testing on 3/4 inch PAX pipe in air with no insulation, and ¾ inch thick foam insulation (R-4.7 hr ft^2 F/Btu). Those results show that UA is a function of both the temperature difference between the pipe and the surrounding air and the flow rate. In general UA values are higher at higher temperature differences, meaning that pipe heat loss is greater at higher temperature differences in a non-linear manner. For practical engineering calculations, however, we can curve-fit the highest UA values observed (generally the higher temperature difference cases), and use those values for most calculations as if they were independent of temperature difference. Figure 7-1 shows scatter-plots of all the data points for the various UA values vs flow rate for all the tests, plus curve-fit lines through the highest UA values vs flow rate for each insulation level. Note that variation of UA with temperature difference is most pronounced for bare pipe, and is much smaller for insulated pipes. This is probably because the actual pipe surface temperature for heat transfer varies more with flow rate in the bare pipe case than do the insulation external surface temperatures. . Note especially that the $UA_{zero-flow}$ for bare pipe varies substantially, and this value should theoretically not be a function of temperature difference between the water in the pipe and the air, because each cool-down test sees continuously dropping pipe temperature, and hence continuously dropping temperature difference. One therefore has substantial latitude in what value one uses to curve-fit the $UA_{zero-flow}$ data point for the bare pipe case.

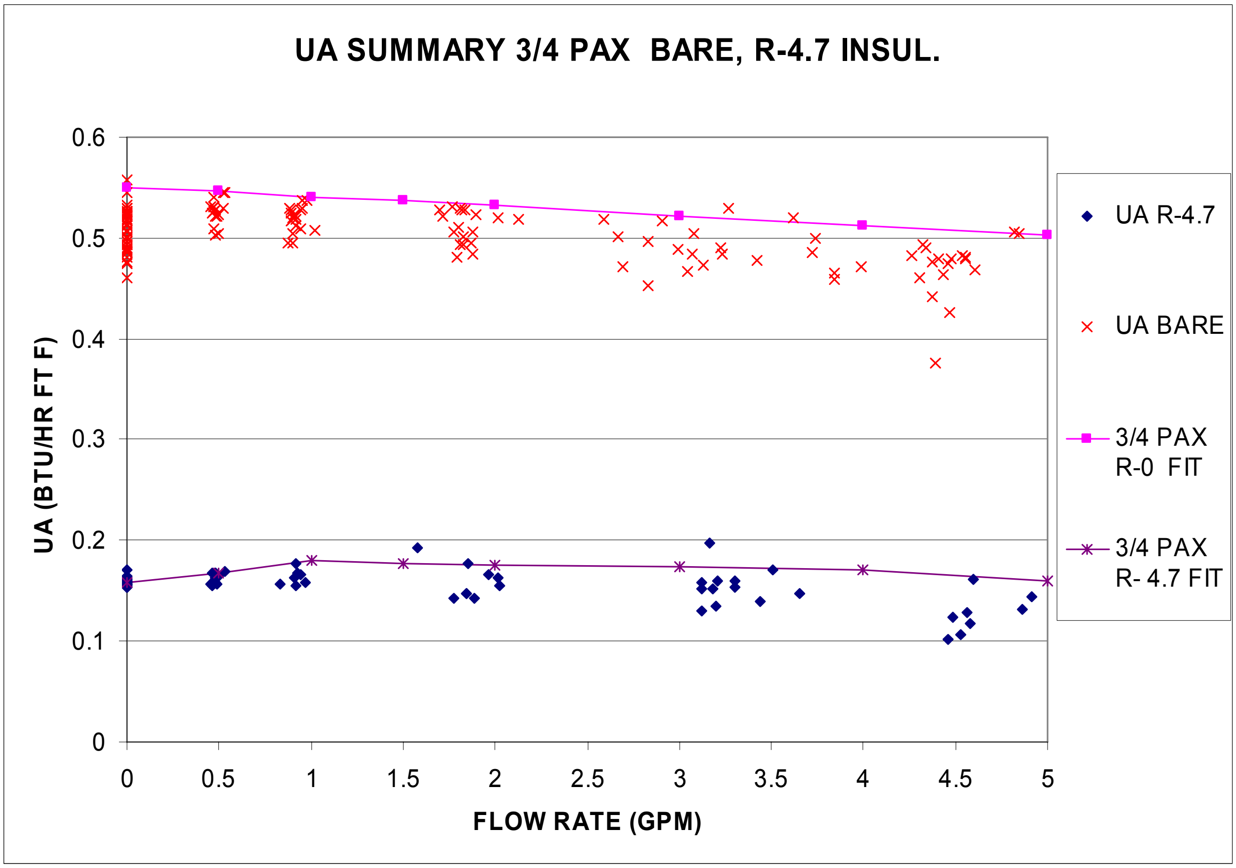


Figure 7-1
Measured Pipe Heat Loss UA Values vs Flow Rate for 3/4 Inch PAX Pipe

Figure 7-1 shows that for 3/4 inch PAX pipe, UA values at zero flow are generally somewhat lower than for cases where there is flow, although the "highest-value" curve fit line for the uninsulated case doesn't quite follow this trend.

Figure 7-1 also reveals an unexpected difference in the shape of the UA curves vs flow rate compared to the rigid copper pipes tested. With the rigid copper pipes, UA increased with increasing flow rate and appeared to reach an asymptotic value at higher flow rates. The ¾ PAX piping reveals a trend of decreasing UA with increasing flow rate. It is believed this behavior is due to an increasing length of slip-flow region in the beginning of the pipe as flow rate increases. Slip-flow appears to reduce heat transfer rate compared to fully developed or normally-developing flow. As flow rate increases, the greater length of the slip-flow region, with its reduced heat transfer, impacts the calculated overall heat transfer rate for the entire pipe length on which the calculation is based. See section 8 of this report comparing UA results from all tests for more discussion.

Delivery-Phase AF/PV Ratio

Section 4 contains a discussion of the general types of behaviors of AF/PV ratio that were seen during testing, so that discussion will not be repeated here. Appendix E gives complete data summary tables for all valid tests of this configuration. This section focuses on presenting quantitative summary results. An important general observation about ¾ PAX piping is that the slip-flow phenomenon described in section 4 develops quite readily; much more easily than in the rigid copper pipes tested. The result is that slip-flow exists over a much broader range of conditions than in the rigid copper pipe. When slip-flow occurs, flow is essentially perfect plug-flow with almost no mixing or degradation of temperature of the hot water throughout the slip-flow region of the pipe. Moreover, slip-slow was observed to sometimes exist over long pipe-lengths, including sometimes the entire length of the test section.

3/4 Inch Bare PAX Pipe AF/PV Results

Figures 7-2 and 7-3 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the 3/4 inch bare PAX pipe tests. Figure 7-2 shows results for a pipe length of 25.3 ft (the first U-bend), corresponding to 0.6697 gallons pipe volume. Figure 7-3 shows results for a pipe length of 100.2 ft – the end of the entire test section, corresponding to 2.6557 gallons pipe volume. Figures 7-4 and 7-5 show AF/PV ratios vs pipe length, cross-plotted against TDR for two different nominal flow rates – 0.5 and 4.0 gpm respectively.

The individual plots in figures 7-2 through 7-5 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.163 – 0.196) and lower lines corresponding to the highest TDRs tested (nominally 0.703 – 0.723). Note, however, that the lines for the higher TDR ratios have many AF/PV values of 1.0, so they do not show up on the plot. This is because flow in the ¾ PAX piping is often in the slip-flow regime. The lines proceed from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

Examining figure 7-2, we see that for this pipe configuration, AF/PV ratio was high in the entrance region at low flow rates, and decreased with increasing flow rate, especially at low TDR. However, unlike for the rigid copper pipes, there was no leveling off of AF/PV. It appears that even at low TDRs, slip-flow occurs in ¾ inch PAX piping at higher flow rates. Moreover, at higher TDRs (greater than about TDR = 0.5, flow in the entrance region is slip-flow, even at the lowest flow rates tested. In figure 7-2, lines indicating TDRs greater than 0.5 are not visible, because entrance region AF/PV equals 1.0 at all flow rates. Behavior corresponding to region B as discussed in Figures 4-2 and 4-3, where flow stratification and pipe axial heat conduction have a detrimental effect on AF/PV ratio does not appear to exist, or at best exists in only a limited fashion, in ¾ inch PAX piping. This is probably because the lower surface adhesion of the plastic pipe and its tendency to develop slip-flow prevent the stratification phenomenon.

It is worth noting that slip-flow is a non-steady-state phenomenon. As such, it does not always repeatedly occur under the same conditions. Rather, slip-flow behaves more-or-less generally the same, within uncertainty bounds. This uncertain behavior causes the relative position of some of the plots in figures 7-2 through 7-7 to not all be in strict order from top-to-bottom with respect to TDR value and flow rate.

Examining figure 7-3 (long pipe length), we see that slip-flow still exists over the entire length of the test section at high TDRs and high flow rates. We note also, that at low TDRs and low flow rates, in several instances AF/PV went to infinity (hence the vertical lines). This signifies that due to heat transfer to ambient, temperature of delivered water dropped below 105 F at the outlet of the test section during the test. The vertical line at approximately 2 gpm flow rate is misleading because there was no valid 1 gpm data point in that test sequence. In reality, the vertical line would be from roughly the 1 gpm location instead of 2 gpm (i.e. the infinite value occurs at a flow rate below 1 gpm). The vertical lines on figures 7-2 through 7-5 correspond to region A as described in figures 4-2 and 4-3, where heat transfer to ambient is an important effect. If the ¾ inch rigid copper piping would have been 100 feet long instead of 86 feet, lines in figures 5-3 and 5-4 would have gone vertical at low TDRs and longer pipe lengths as well.

Examining figure 7-3, we see again that there appears to be little or no region B as shown on figures 4-2 and 4-3, where stratification and axial heat conduction have a detrimental effect on AF/PV ratio. Moreover, it appears that the effects of slip-flow start to be measurable at TDRs above about 0.35 at the lowest flow rates (0.5 gpm) tested. Examining figure 7-4, we see that at high flow rates, slip-flow has a measurable impact on AF/PV ratio, even at the lowest TDRs tested (TDR = 0.175) Moreover, it appears that at higher flow rates, such as the 3.9 – 4.5 gpm cases shown, flow is slip-flow in bare ¾ PAX piping, for essentially the entire length of the 100 ft long test section, at essentially all TDR > 0.35, causing AF/PV ratios to equal 1.0 at all pipe lengths under those conditions.

The flow rates at which slip-flow can develop are a function of TDR, as is the pipe length over which the slip-flow will persist. Whether or not slip-flow will develop, and at what flow rates and TDRs, and over what pipe length the slip-flow will persist are also functions of the pipe diameter, smoothness of the inner pipe wall, presence or absence of fittings that can cause turbulence, and the pipe material (affects surface adhesion and hence shear resistance at the water/wall interface).

Figures 7-6 and 7-7 show all AF/PV results from all tests on bare 3/4 inch PAX piping, plotted on the same plots. In figure 7-6 upper lines correspond to lower TDRs while lower lines correspond to higher TDRs. In figure 7-6 many of the high TDR cases are not visible because their AF/PV ratio is 1.0 over the entire flow rate range. In figure 7-7 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. Many of the high flow rate, high TDR cases are not visible on figure 7-7 because their AF/PV ratio is 1.0 at all pipe lengths due to slip-flow. The complete summary data tables included in Appendix E allow the reader to reconstruct plots isolated by any of the important parameters. We can see from figures 7-6 and 7-7 that for ¾ inch bare PAX piping, AF/PV ratios normally range from a low of around 1.0 to a high of around 2.0, with most in the range of 1.0 to 1.5.

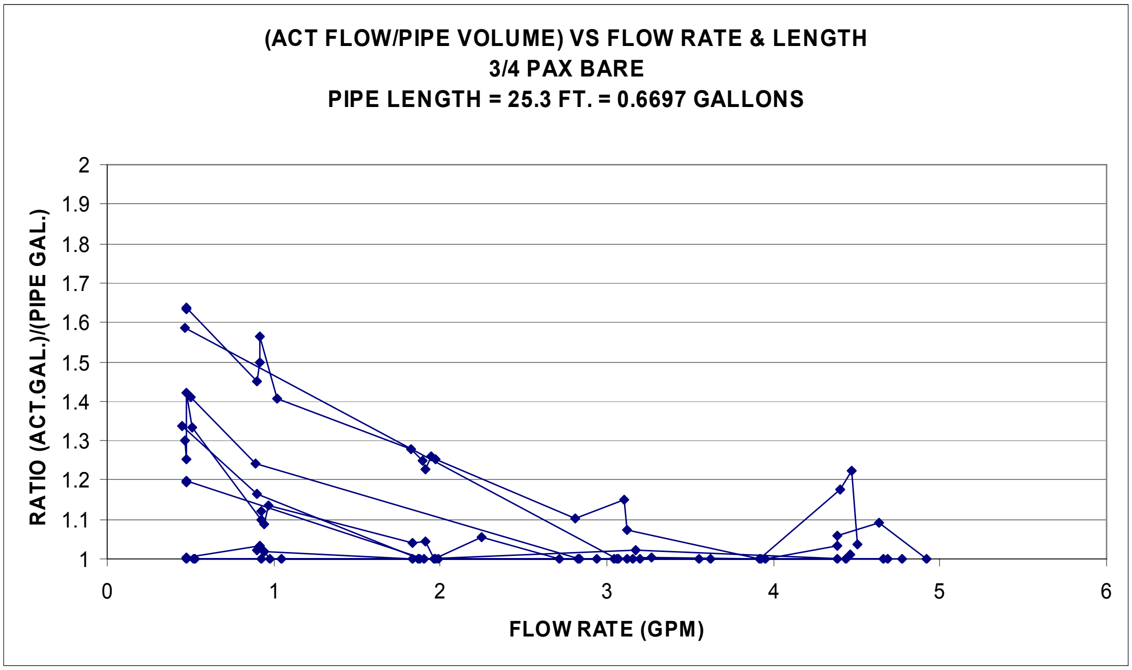


Figure 7-2
Measured AF/PV Ratio vs Flow Rate & TDR For Bare 3/4 Inch PAX Pipe – 25.3 Ft Pipe Length (Upper Plot TDR = 0.163 – 0.196, Lower Plot TDR = 0.453 – 0.487 Above TDR = 0.5, AF/PV = 1.0 at all flow rates)

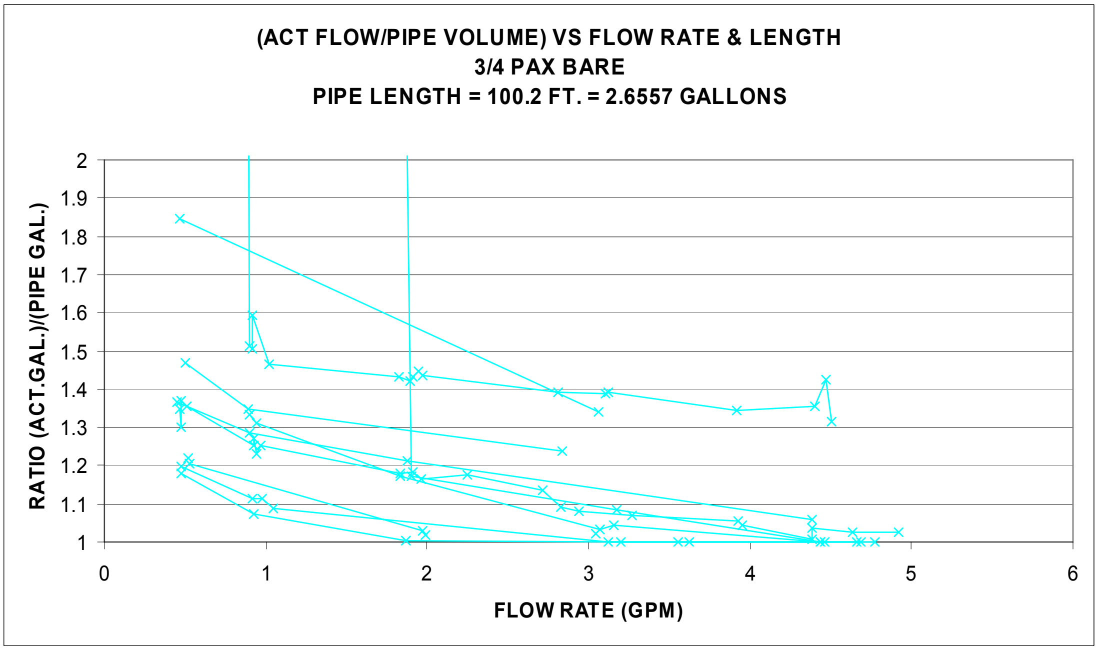


Figure 7-3
Measured AF/PV Ratio vs Flow Rate & TDR For Bare 3/4 Inch PAX Pipe – 100.2 Ft Pipe Length (Upper Plot TDR = 0.163 – 0.196, Lower Plot TDR = 0.711– 0.723

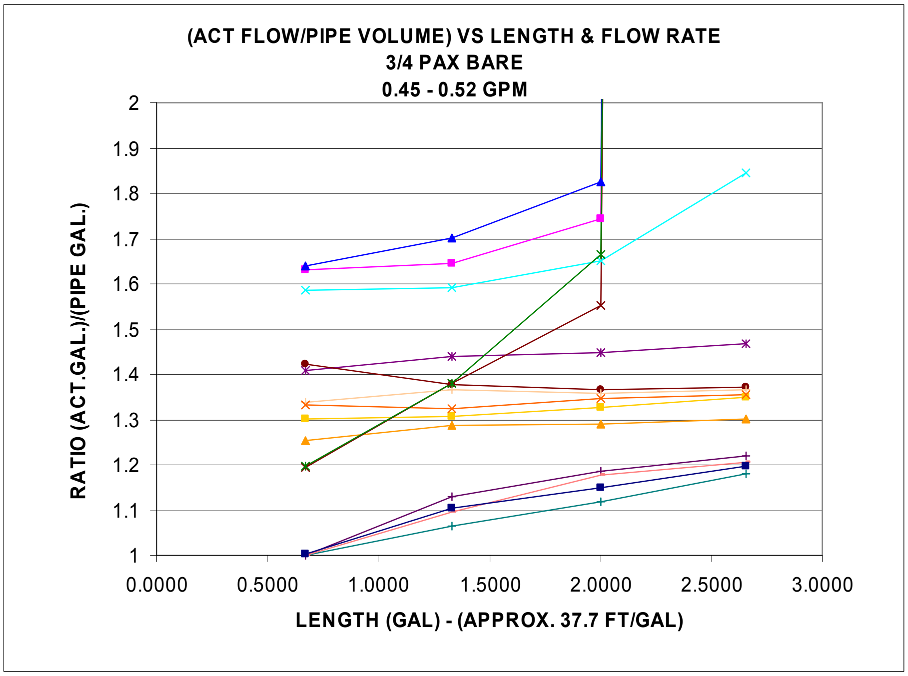


Figure 7-4
Measured AF/PV Ratio vs Pipe Length & TDR For Bare 3/4 Inch PAX Pipe – 0.45 -0.52 GPM Flow Rate (Upper Plot TDR = 0.175, Lower Plot TDR = 0.723)

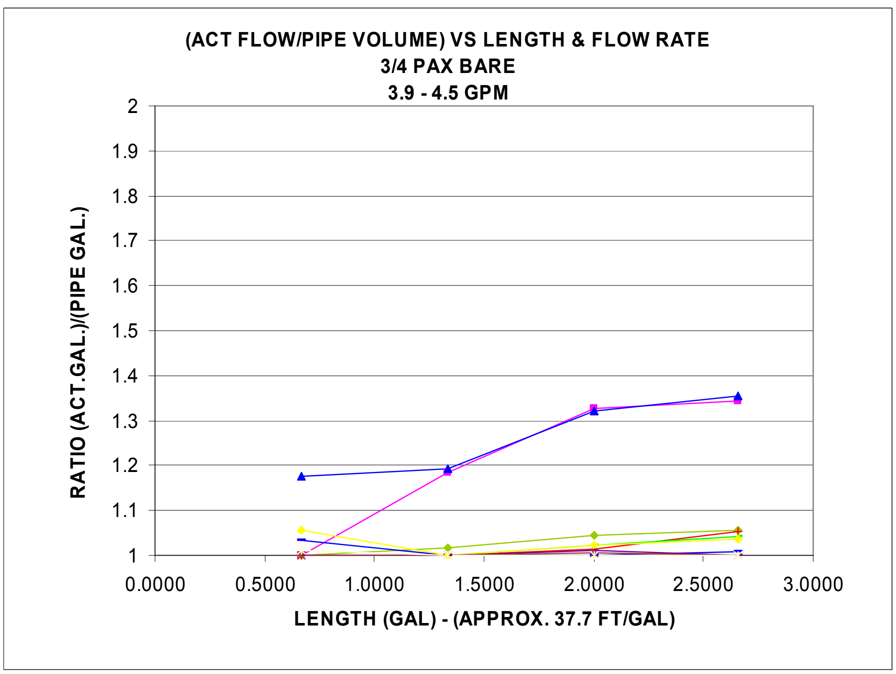


Figure 7-5
Measured AF/PV Ratio vs Length & TDR For Bare 3/4 Inch PAX Pipe – 3.9 -4.5 GPM Flow Rate (Upper Plot TDR = 0.175, Lower Plot TDR = 0.344. Above TDR ≈ 0.35, all AF/PV ≈ 1.0)

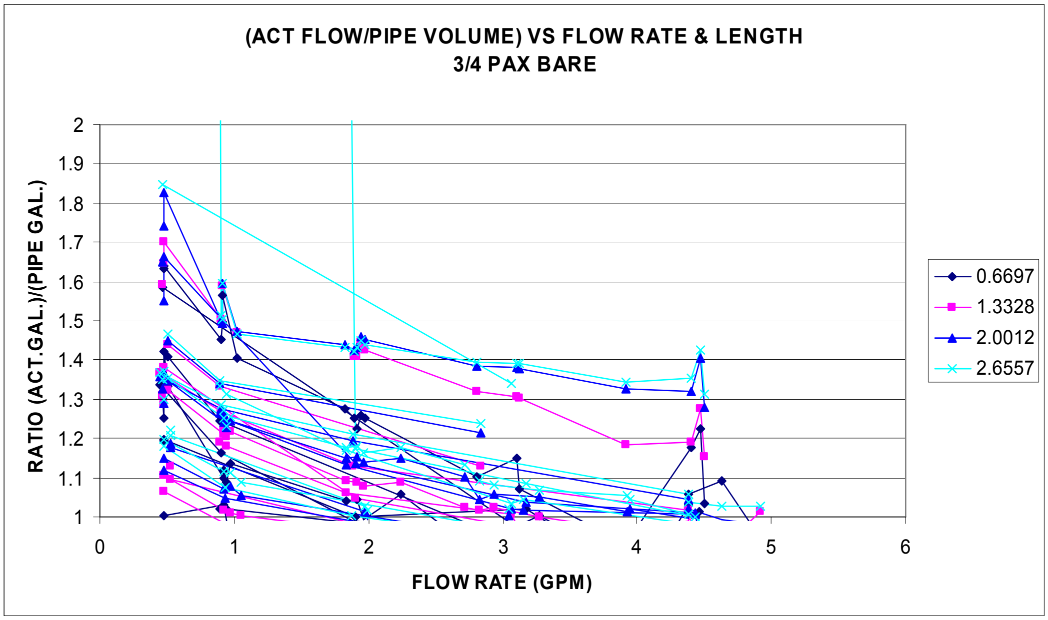


Figure 7-6
Measured AF/PV Ratio vs Flow, Length & TDR For Bare 3/4 Inch PAX Pipe – All Lengths and TDRs (Legend length in gallons, 37.7 ft = 1 gallon) (Many AF/PV = 1.0 & not visible)

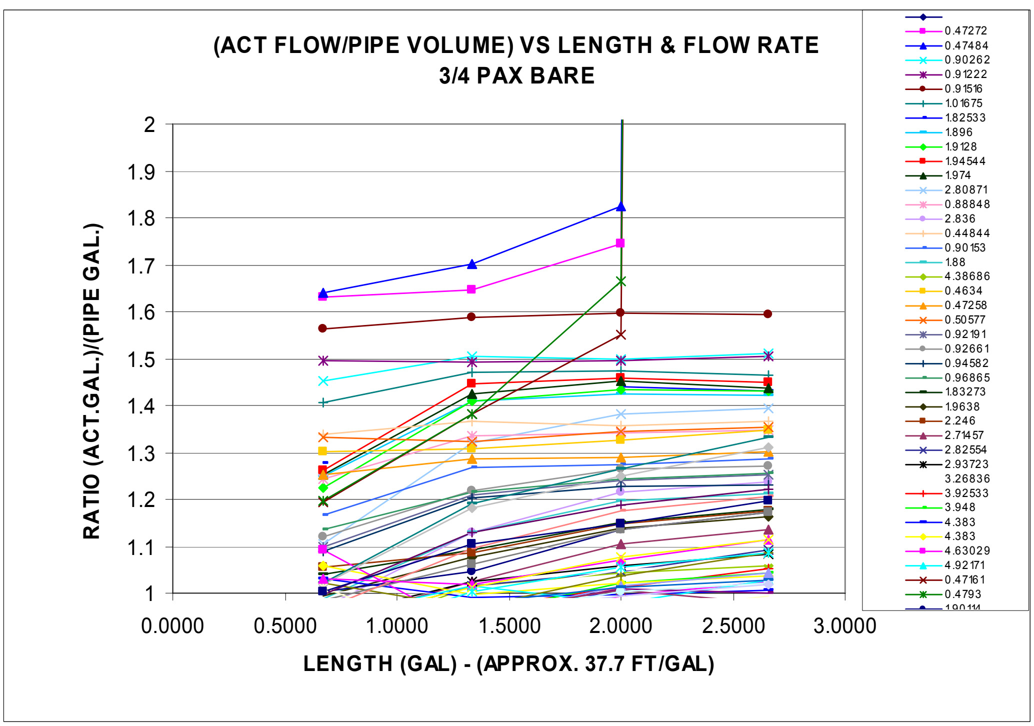


Figure 7-7
Measured AF/PV Ratio vs Pipe Length, Flow Rate & TDR For Bare 3/4 Inch PAX Pipe – All Flow Rates and TDRs (Legend is flow rate in GPM) (Many AF/PV = 1.0 & hence not visible.)

3/4 Inch PAX Pipe With ¾ Inch Thick Foam Insulation AF/PV Results

The foam pipe insulation used for these tests was of the same type most commonly seen in the field site visits - black closed-cell polyethylene or polyolefin foam. Properties for these two materials are similar. R-values were printed on the insulation, like was observed at the field sites. The typical thermal conductivity listed by various manufacturers for these types of pipe insulation was around k = 0.02 Btu/hr ft F. The R-values ranged as shown in table 3-2 for the different foam thicknesses and different pipe sizes, because R-value is based on the outer diameter of the foam. The above listed thermal conductivity can be used in standard heat transfer calculations for multi-layer cylindrical objects (given in most introductory heat transfer textbooks) to calculate the R-value for insulation of a given thickness applied to pipe of a given diameter. The R-value of the 3/4inch thick foam on the 3/4 inch nominal PAX pipe size (approximately 1 inch OD) was R-4.7 hr ft^2F/Btu. The pipe outer diameter was slightly larger than the insulation inner diameter in this case. This was not a problem because the foam pipe insulation is flexible, and could be stretched in order to close the longitudinal seam.

Figures 7-8 and 7-9 show AF/PV ratios vs flow rate, cross-plotted against TDR for two different pipe lengths in the tests on 3/4 inch PAX pipe with R-4.7 insulation. Figure 7-8 shows results for a pipe length of 25.31 ft (the first U-bend), corresponding to 0.6697 gallons pipe volume. Figure 7-9 shows results for a pipe length of 100.2 ft – the end of the entire test section, corresponding to 2.6557 gallons pipe volume. Figures 7-10 and 7-11 show AF/PV ratios vs pipe length, cross-plotted against TDR for 0.5 and 4.5 gpm flow rates respectively.

The individual plots in figures 7-8 through 7-11 correspond to different TDRs, with upper lines corresponding to the lowest TDRs tested (nominally 0.173 – 0.2) and lower lines corresponding to the highest TDRs tested (nominally 0.652 – 0.673). Note, however, that the lines for the higher TDR ratios have many AF/PV values of 1.0, so they do not show up on the plots. This is because flow in the ¾ PAX piping is often in the slip-flow regime. The lines proceed generally from lowest to highest TDR from top to bottom, but in a non-linear manner. Development of correlations relating AF/PV ratio vs flow rate, pipe length, and TDR was beyond the scope of the current effort, and should be done after tests on more piping configurations are completed.

The behavior of ¾ PAX piping with R-4.7 insulation is very similar to bare ¾ PAX piping, except for the lack of significant effects due to heat loss to ambient at low flow rates and low TDRs. Hence region A as shown in figures 4-2 and 4-3 does not appear to exist. It appears that slip-flow develops just as readily in ¾ PAX piping without insulation as it does with insulation in the in-air environment. The insulation reduces heat loss to ambient that can negatively impact AF/PV ratio at low flow rates and TDR, but does not otherwise significantly affect AF/PV ratio.

Figures 7-12 and 7-13 show all AF/PV results from all tests on 3/4 inch PAX piping with R-4.7 insulation, plotted on the same plots. In figure 7-12 upper lines correspond to lower TDRs, while lower lines correspond to higher TDRs. In figure 7-12 many of the high TDR cases are not visible because their AF/PV ratio is 1.0 over the entire flow rate range. In figure 7-13 upper lines correspond to lower TDRs and lower flow rates, while lower lines correspond to higher TDRs and higher flow rates. Many of the high flow rate, high TDR cases are not visible on figure 7-7 because their AF/PV ratio is 1.0 at all pipe lengths due to slip-flow. The complete summary data tables included in Appendix E allow the reader to reconstruct plots isolated by any of the important parameters. We see from figures 7-12 and 7-13 that for 3/4 inch PAX piping with R-4.7 insulation, AF/PV ratios mostly behaved similarly to the bare pipe case, with normal AF/PV ratios being in the range of 1.0 to 2.0, and most being in the range of 1.0 to 1.5.

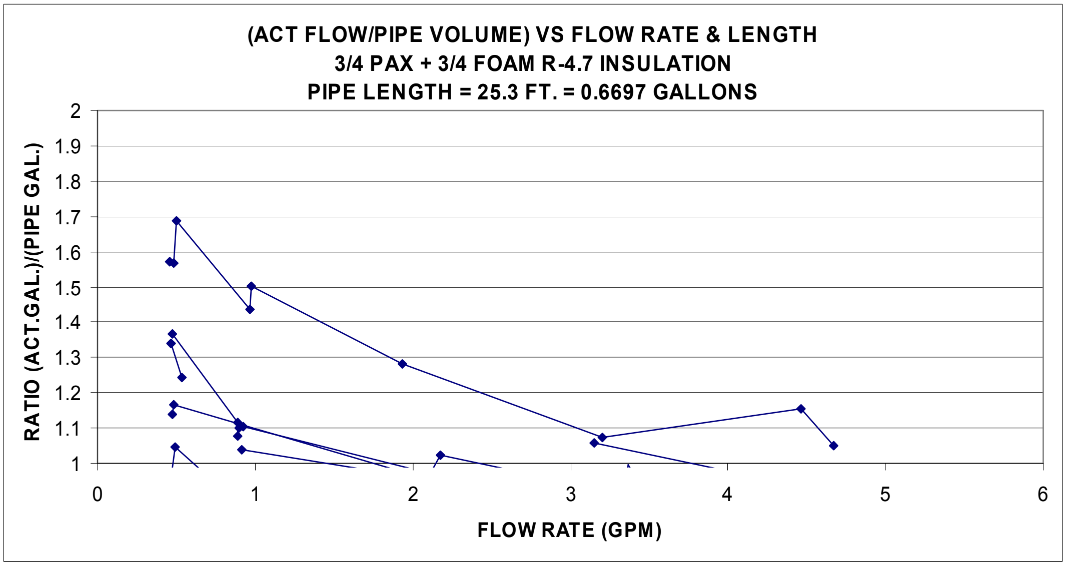


Figure 7-8
Measured AF/PV Ratio vs Flow Rate & TDR For 3/4 Inch PAX Pipe With R-4.7 Insulation– 25.3 Ft Pipe Length (Upper Plot TDR = 0.173 – 0.199, Lower Plot TDR = 0.454 – 0.472, Plots for TDR>0.5 not visible since AF/PV ≈ 1.0 at all flow rates)

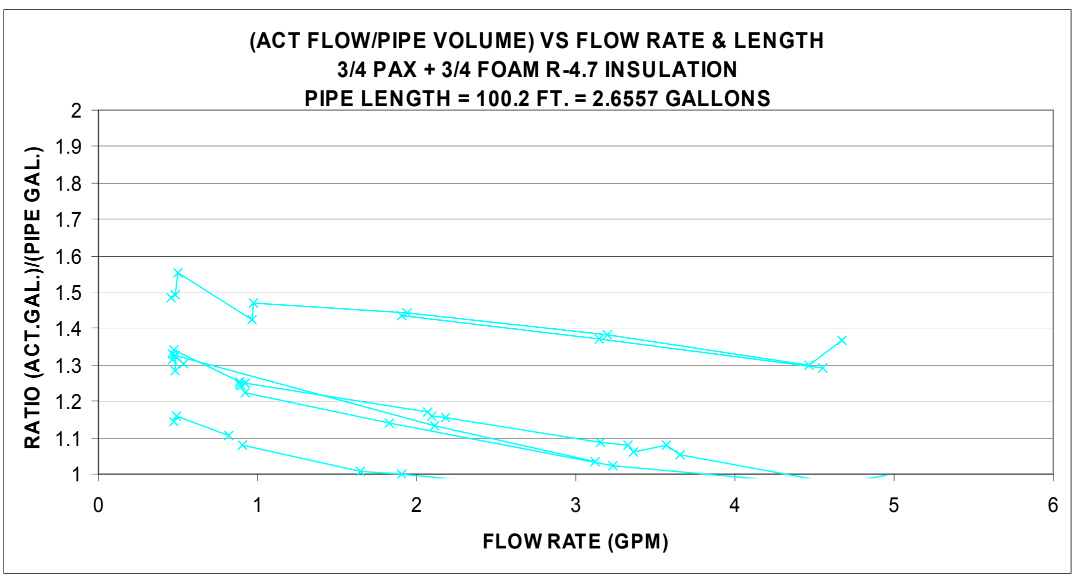


Figure 7-9
Measured AF/PV Ratio vs Flow Rate & TDR For 3/4 Inch PAX Pipe With R-4.7 Insulation– 100.2 Ft Pipe Length (Upper Plot TDR = 0.173 – 0.199, Lower Plot TDR = 0.652 – 0.673)

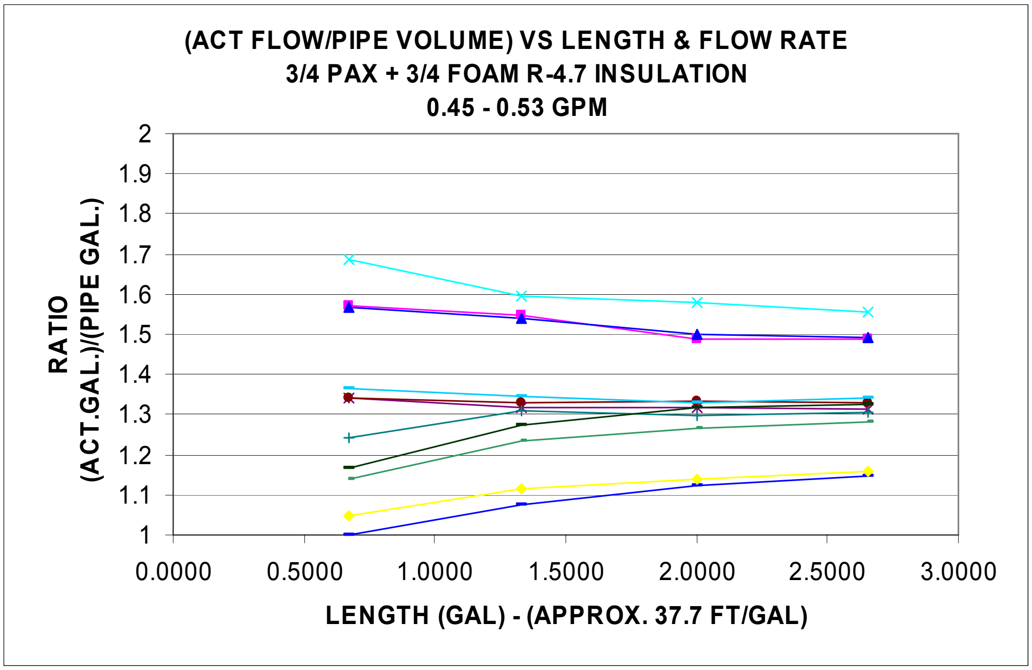


Figure 7-10
Measured AF/PV Ratio vs Pipe Length & TDR For 3/4 Inch PAX Pipe With R-4.7 Insulation– 0.45 – 0.53 GPM (Upper Plot TDR = 0.173, Lower Plot TDR = 0.647)

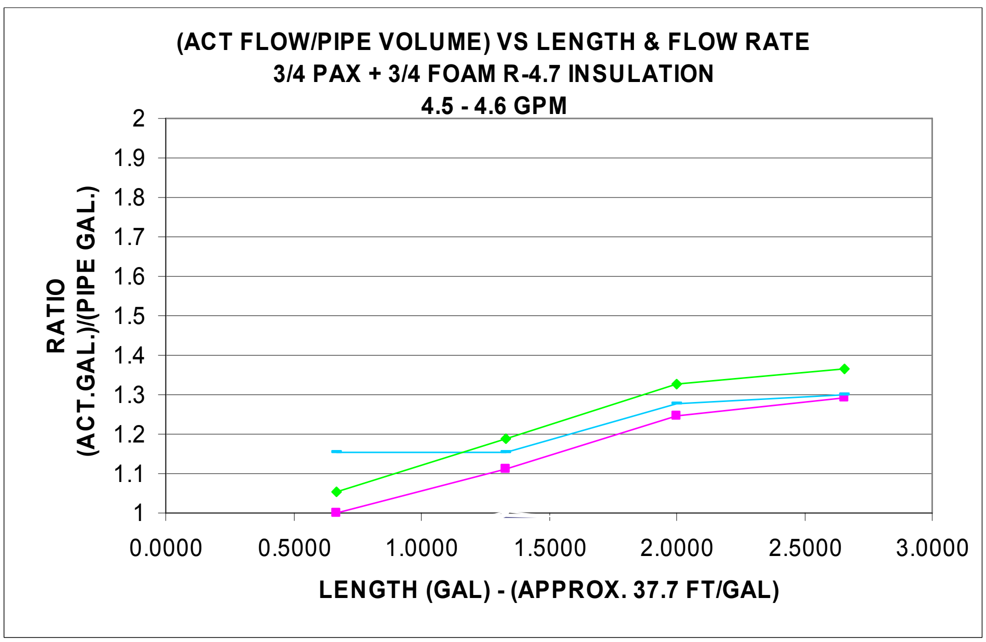


Figure 7-11
Measured AF/PV Ratio vs Pipe Length & TDR For 3/4 Inch PAX Pipe With R-4.7 Insulation– 4.5 – 4.6 GPM (Upper Plot TDR = 0.178, Lower Plot TDR = 0.203, All plots for TDR>0.35 not visible because AF/PV ≈1.0 at all pipe lengths at this flow rate)

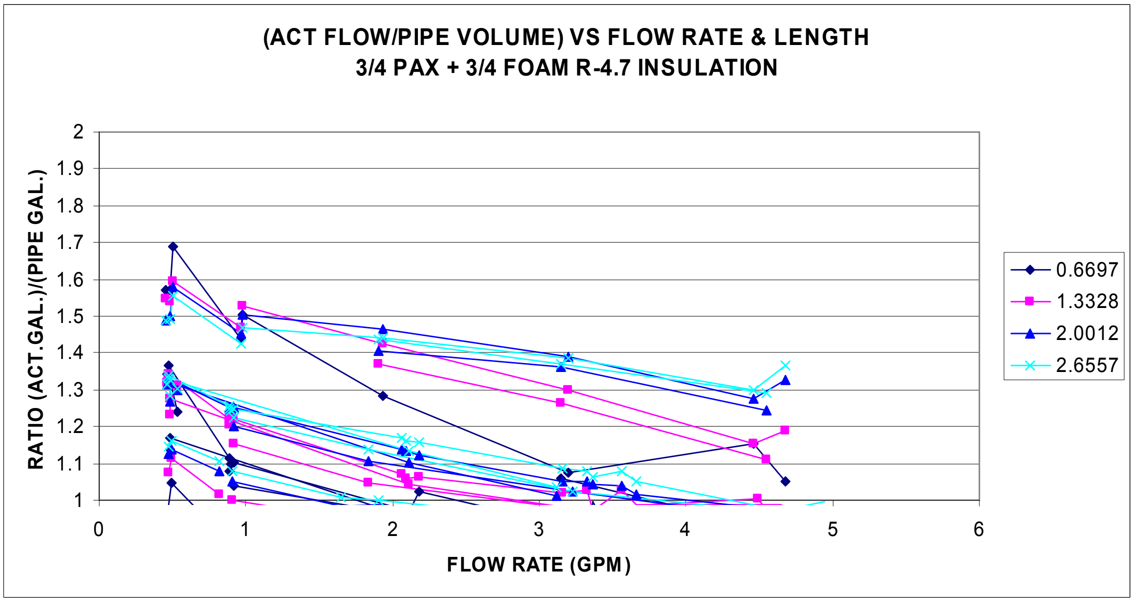


Figure 7-12
Measured AF/PV Ratio vs Flow, Length & TDR For 3/4 Inch PAX With R-4.7 Insul.– All Lengths and TDRs (Legend length in gal., 37.7 ft = 1 gal.)(Many AF/PV = 1.0 & not visible)

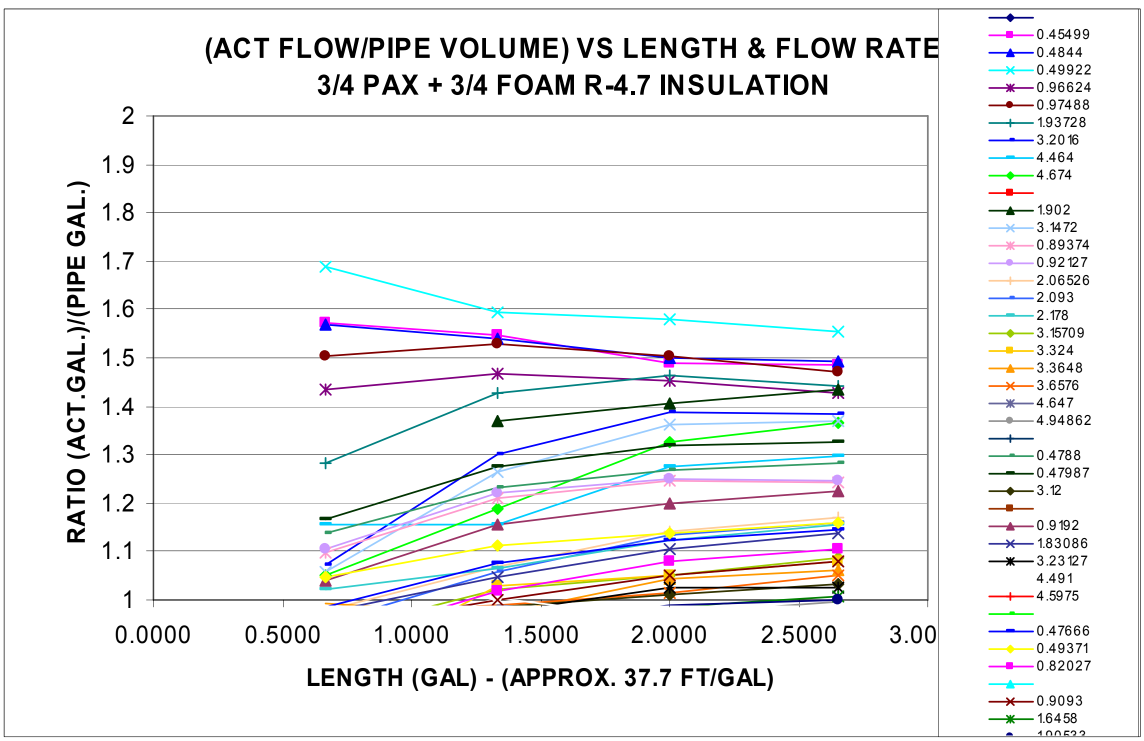


Figure 7-13
Measured AF/PV Ratio vs Length, Flow & TDR For 3/4 Inch PAX Pipe With R-4.7 Insul.– All Flow Rates and TDRs (Legend is flow rate in GPM) (Many AF/PV = 1.0 & hence not visible)

Comparing AF/PF Results for Bare and R-4.7 Insulation on 3/4 Inch PAX Pipe

Behavior trends for the ¾ inch PAX piping were similar to those for the ¾ inch and ½ inch rigid copper piping.

While there was a large and readily apparent difference in the pipe heat loss rate (UA value) on insulated vs uninsulated pipe, the impact of insulation on AF/PV ratios and hence water and energy waste) during the hot water delivery phase was less pronounced. For the most part, AF/PV ratios were nearly the same for insulated vs uninsulated ¾ inch PAX pipe. There were, however, some important distinctions.

One important difference was that for the insulated configurations, water exit temperatures at low flow rates remained well above 105 F, indicating that AF/PV ratios would not go to infinity under normally occurring design and operation conditions, unlike for bare pipe. That is to say, the region A behavior shown in figures 4-2 and 4-3, where heat transfer to ambient significantly affected AF/PV ratio, did not exist for the insulated pipe cases. Heat loss rates were so high in the bare pipe configuration, even for pipes in still air, that for low flow rates it would sometimes be necessary to increase tank setpoint temperature in order to obtain "hot enough (105 F) water from pipes of lengths that would often be encountered in practice. Insulated pipes lost much less heat. In fact, one can use the measured pipe UA values to calculate the critical pipe length beyond which 105 F water cannot be obtained at any selected flow rate. This is done in section 8 of this report, comparing the UA values from all test configurations.

Unlike with rigid copper pipe, adding insulation did not appear to significantly increase the tendency for flow to go into the slip-flow regime. Apparently, in PAX pipe inner surface conditions are favorable for slip-flow whether or not the pipe is insulated. As noted in section 4, slip-flow reduces temperature degradation in the hot water traveling to the fixtures, thus reducing water and energy waste during the delivery phase of hot water flow.

8
PIPE HEAT LOSS RATE COMPARISONS – HORIZONTAL PIPES IN AIR

UA Value Summaries

Figure 8-1 shows curve-fits to peak measured UA values vs flow rate for all of the horizontal in-air piping configurations tested in this phase I effort. In general we can say that UA value at zero flow is essentially the same or lower (and sometime substantially lower) than the UA when there is water flowing in the pipes. Bare ¾ PAX piping is a slight exception to this rule. Note, however, that as noted in sections 5, 6, and 7, one has substantial latitude in selecting which $UA_{zero\text{-}flow}$ value to use for bare pipes. The $UA_{flowing}$ values for bare pipe fall in roughly the same ratios as the external surface areas of the pipes, for cases when water is flowing in the pipes. The nominal ¾ inch PAX piping has both a larger inner and outer diameter than does nominal ¾ inch rigid copper pipe – see appendices C and E. This means that when water is flowing in the pipes, overall U values based on pipe external surface area fall in a close range for the three pipes tested. Using the curve-fit UA values for bare pipe at 5.0 gpm, and the external surface areas of the three pipes tested, we calculate bare pipe U values of approximately 2.2 Btu/hr ft^2 For the ½ inch rigid copper pipe, and 1.92 Btu/hr ft^2 F for both the ¾ inch rigid copper and the ¾ inch PAX piping. However, it appears that heat loss from bare ¾ inch PAX piping is proportionately higher than for ¾ copper pipe at low water flow rates for unknown reasons. The values of U estimated using the method above can probably be used to estimate UA of similar types of bare pipes (i.e. copper or PAX) that are one size larger or smaller than those tested. Error in calculated heat loss using this method would probably be on the order of about 15%. This error estimate is based on the differences we calculate in U value for ½ inch vs ¾ inch pipe.

It appears that R-4.7 insulation on PAX pipe is more effective than on ¾ rigid copper at higher water flow rates. Both the percentage reduction in heat loss rate and the absolute value of heat loss rate (UA factor) are lower when ¾ in thick R-4.7 foam is applied to ¾ inch PAX as compared to ¾ inch rigid copper pipe. However, at zero and low flow, UA values of ¾ inch PAX and ¾ inch rigid copper pipe are similar when R-4.7 foam insulation is applied. The difference in flowing vs non-flowing performance appears related to the fact that slip-flow was occurring in the PAX piping and mostly not in the rigid copper.

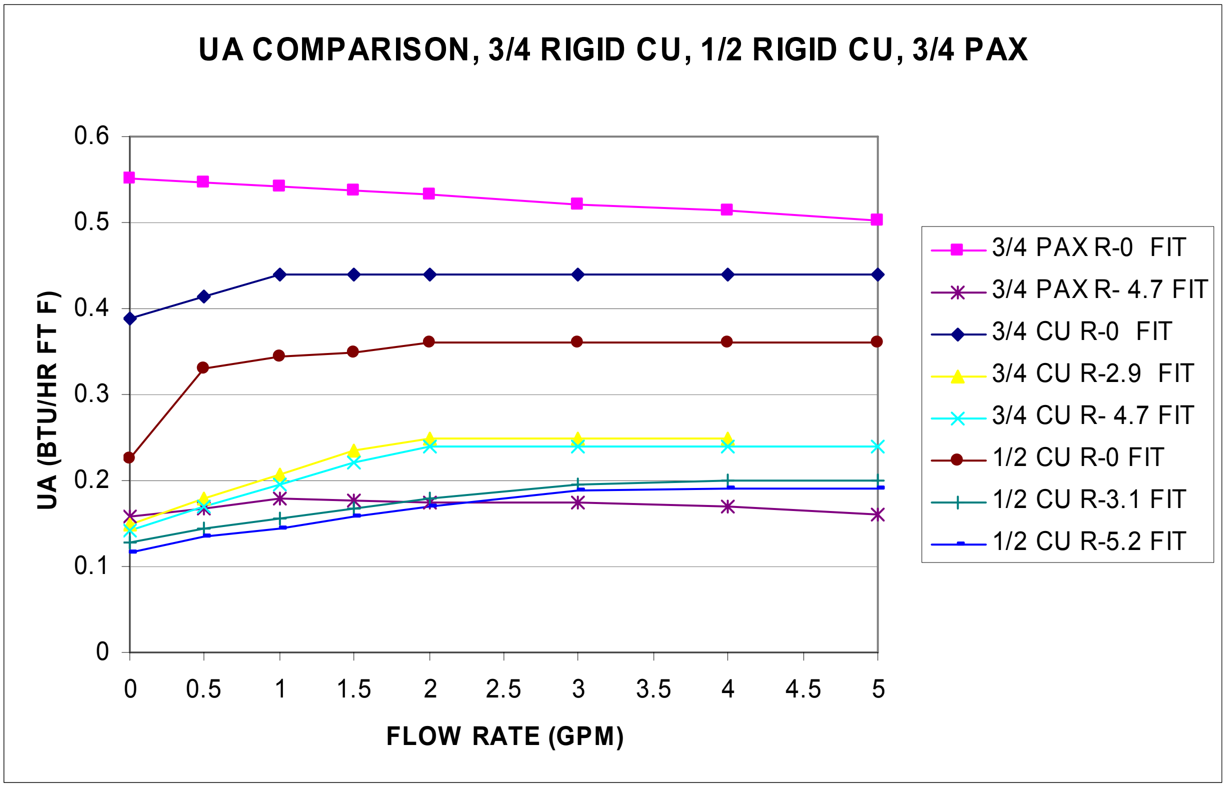


Figure 8-1
Measured Pipe Heat Loss UA Values vs Flow Rate for All Pipe Configurations Tested

If computerized calculations are performed, the UA values shown in figure 8-1 can be entered as functions of flow rate. Tables of data points for the curve fits of figure 8-1 are given in appendices C, D, and E. For simpler hand-calculations, it is possible to create constant estimates of pipe UA value for the various pipe configurations tested, based on peak UA values observed. Separate constant values should be used for zero-flow (cool-down) vs flowing conditions. Table 8-1 shows the constant values that should be used.

Table 8-1
Pipe UA Value Summary

NOMINAL PIPE SIZE (IN.)	FOAM INSUL. THICK. (IN.)	ZERO FLOW UA (BTU/HR FT F)	HIGH-VALUE UA (BTU/HR FT F)
½ Rigid CU	0	.226	.36
½ Rigid CU	½ (R-3.1)	.128	.20
½ Rigid CU	¾ (R-5.2)	.116	.19
¾ Rigid CU	0	.388	.44
¾ Rigid CU	½ (R-2.9)	.150	.25
¾ Rigid CU	¾ (R-4.7)	.142	.24
¾ PAX	0	.55	.546
¾ PAX	¾ (R-4.7)	.158	.18

The data in figure 8-1 can be used to develop a significant amount of useful derivative information. Examples include:

- Estimate pipe cool-down rates and temperature vs time in cool-down
- Estimate temperature drop through piping under flowing conditions
- Estimate pipe heat loss rate for typical conditions
- Calculate critical pipe length beyond which all useful heat energy is lost vs flow rate and temperature conditions
- Calculate percentage of energy put into heating the water that is lost to ambient vs flow rate, pipe length, and temperature conditions

Pipe Cool-Down Rates

If one knows, or can calculate the effective thermal mass of piping, water, and insulation, one can use the zero-flow UA values from table 8-1 to calculate time to cool down to a selected temperature, starting from a given initial temperature, under a given ambient air temperature condition. Similarly, one can calculate pipe temperature at any point in time during the cool-down. This is done using the exponential temperature decay function discussed in Appendix B. Table 8-2 shows times for the various pipe configurations to cool-down to 105 F, starting from several different initial pipe temperatures, at an assumed ambient air temperature condition corresponding to the U.S. Department of Energy (DOE) water heater test conditions of 67.5 F. The pipe thermal mass values used in these calculations are given in Appendices C, D, and E.

Table 8-2
Pipe Cool-Down Times (Minutes) to Reach 105 F (Tair = 67. 5 F)

NOM. PIPE SIZE (IN.)	FOAM INSUL. THICK. (IN.)	T hot 135 F	T hot 130 F	T hot 125 F	T hot 120 F	T hot 115 F	T hot 110 F
½ Rigid CU	0	19.8	17.20	14.4	11.3	7.9	4.2
½ Rigid CU	½ (R-3.1)	35.8	31.1	26.0	20.50	14.4	7.6
½ Rigid CU	¾ (R-5.2)	40.4	35.1	29.4	23.1	16.2	8.6
¾ Rigid CU	0	22.74	19.7	16.5	13.0	9.1	4.8
¾ Rigid CU	½ (R-2.9)	59.8	51.9	43.5	34.2	24.0	12.7
¾ Rigid CU	¾ (R-4.7)	64.0	55.6	46.5	36.6	25.7	13.6
¾ PAX	0	18.1	15.8	13.2	10.4	7.3	3.9
¾ PAX	¾ (R-4.7)	64.8	56.3	47.1	37.1	26.1	13.8

Pipe Steady-State Temperature Drop

The peak UA values from figure 8-1 can be used to calculate the steady-state drop in temperature as water flows through a length of the various pipes. Table 8-3 shows calculated temperature drop through 100 feet of pipe, assuming an entering hot water temperature of 135 F and an air temperature of 67.5 F, corresponding to the DOE water heater test conditions, for various water flow rates. Table 8-3 uses UA values for each flow rate from figure 8-1, and the equations discussed in Appendix B for calculating $UA_{flowing}$. The temperature drops shown reflect a decreasing rate of temperature drop with length as the water looses temperature while flowing through the pipe. Table 8-4 shows temperature drops starting from an initial hot water temperature of 115 F instead of 135 F.

Table 8-3
Pipe Flow Steady-State Temperature Drop (F) vs Flow Rate (100 Ft. Pipes, Thot = 135 F, Tair = 67. 5 F)

NOM. PIPE SIZE (IN.)	FOAM INSUL. THICK. (IN.)	0.5 GPM	1.0 GPM	1.5 GPM	2.0 GPM	3.0 GPM	4.0 GPM	5.0 GPM
½ Rigid CU	0	8.4	4.5	3.1	2.4	1.6	1.2	0.97
½ Rigid CU	½ (R-3.1)	3.8	2.1	1.5	1.2	0.88	0.67	0.54
½ Rigid CU	¾ (R-5.2)	3.6	1.9	1.4	1.1	0.84	0.64	0.51
¾ Rigid CU	0	10.4	5.7	3.9	2.9	2.0	1.5	1.2
¾ Rigid CU	½ (R-2.9)	4.7	2.8	2.1	1.7	1.1	0.84	0.67
¾ Rigid CU	¾ (R-4.7)	4.5	2.6	2.0	1.6	1.1	0.81	0.65
¾ PAX	0	13.3	6.9	4.7	3.5	2.3	1.7	1.3
¾ PAX	¾ (R-4.7)	4.4	2.4	1.6	1.2	0.78	0.57	0.43

Table 8-4 shows steady-state temperature drop similar to table 8-3, but assuming T_{hot} = 115 F.

Table 8-4
Pipe Flow Steady-State Temperature Drop (F) vs Flow Rate (100 Ft. Pipes, Thot = 115 F, Tair = 67. 5 F)

NOM. PIPE SIZE (IN.)	FOAM INSUL. THICK. (IN.)	0.5 GPM	1.0 GPM	1.5 GPM	2.0 GPM	3.0 GPM	4.0 GPM	5.0 GPM
½ Rigid CU	0	5.9	3.2	2.2	1.7	1.1	0.85	0.68
½ Rigid CU	½ (R-3.1)	2.7	1.5	1.1	0.85	0.62	0.47	0.38
½ Rigid CU	¾ (R-5.2)	2.5	1.4	0.99	0.80	0.59	0.45	0.36
¾ Rigid CU	0	7.3	4.0	2.7	2.1	1.4	1.0	0.83
¾ Rigid CU	½ (R-2.9)	3.3	1.9	1.5	1.2	0.79	0.59	0.47
¾ Rigid CU	¾ (R-4.7)	3.1	1.8	1.4	1.1	0.76	0.57	0.46
¾ PAX	0	9.4	4.9	3.3	2.5	1.6	1.2	0.95
¾ PAX	¾ (R-4.7)	3.1	1.7	1.1	0.83	0.55	0.40	0.30

Pipe Steady-State Heat Loss Rate

The peak UA values from figure 8-1 can also be used to calculate the heat loss rate for steady-state flow of water through a length of the various pipes. Table 8-5 shows calculated heat loss rate from flow through 100 feet of pipe, assuming 135 F entering hot water and 67.5 F air temperatures. Table 8-5 uses UA values for each flow rate from figure 8-1, and the equations discussed in Appendix B for calculating flowing UA. The heat loss rates shown reflect dropping temperature as water flows through the pipe.

Table 8-4
Heat Loss Rate (Btu/hr) for 100 Ft. of Pipe (Thot = 135 F, Tair = 67. 5 F)

NOM. PIPE SIZE (IN.)	FOAM INSUL. THICK. (IN.)	0.5 GPM	1.0 GPM	1.5 GPM	2.0 GPM	3.0 GPM	4.0 GPM	5.0 GPM
½ Rigid CU	0	2086	2250	2308	2387	2401	2408	2413
½ Rigid CU	½ (R-3.1)	951	1030	1121	1204	1307	1343	1345
½ Rigid CU	¾ (R-5.2)	887	965	1049	1138	1261	1276	1278
¾ Rigid CU	0	2580	2843	2884	2905	2927	2937	2944
¾ Rigid CU	½ (R-2.9)	1160	1375	1562	1666	1673	1677	1679
¾ Rigid CU	¾ (R-4.7)	1109	1291	1463	1601	1607	1610	1612
¾ PAX	0	3309	3460	3498	3497	3463	3419	3361
¾ PAX	¾ (R-4.7)	1090	1193	1181	1171	1168	1143	1077

Pipe Heat Loss Critical Length

The peak UA values from figure 8-1 can be used, along with the equations discussed in Appendix B for calculating UA, to calculate the length of pipe it would take for exit temperatures to drop to 105 F, given assumed flow rates and air and entering hot water temperatures. We call this “heat loss critical length” (Critical Length). Critical length is a function of pipe configuration, flow rate, and entering hot water and air temperature, and can be plotted in a variety of ways. Figure 8-2 shows critical length vs flow rate, cross-plotted against entering hot water temperature, for bare ¾ inch PAX piping in 32 F air.

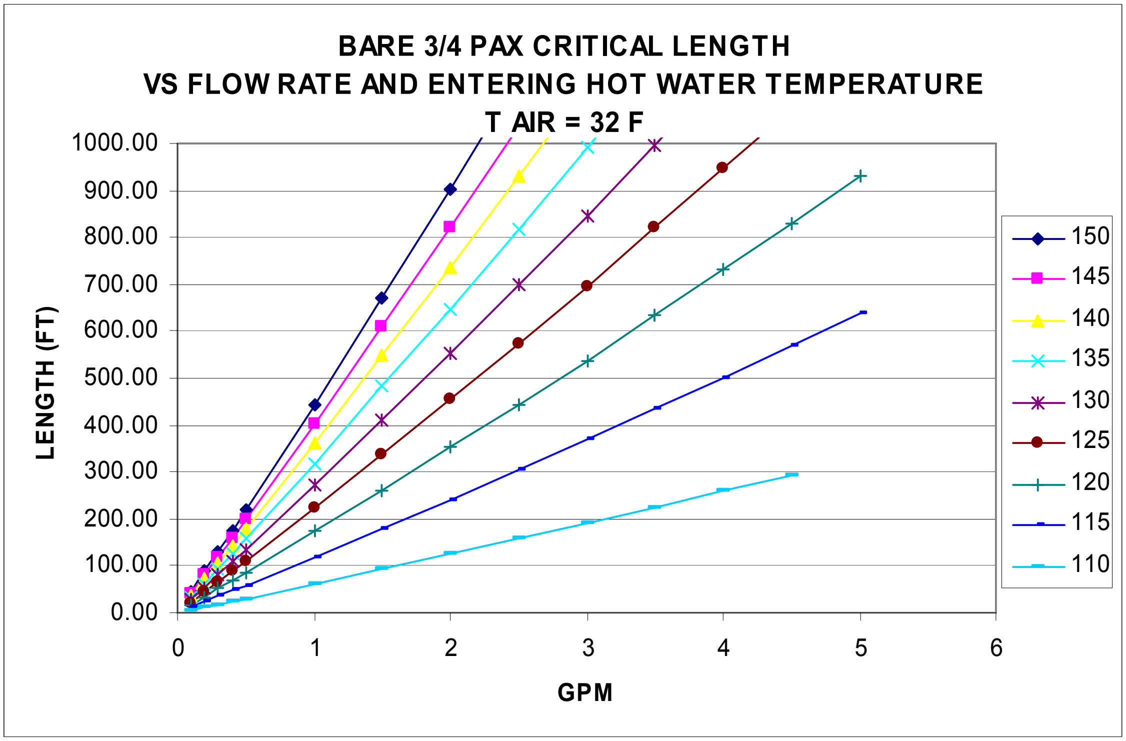


Figure 8-2
Critical Length for Bare ¾ PAX Pipe vs Flow Rate and Temperature (Legend is Entering Hot Water Temperature in F)

Lengths of bare ¾ PAX piping seen in the site visits discussed in Section 3 were often on the order of 100 to 200 feet. (See also the example calculations in appendix A of this report.) Moreover, maximum flow rates at sinks, such as the kitchen sink, are usually on the order of 1 gpm, and such sinks are often used at lower flow rates, such as 0.2 – 0.5 gpm. Examining figure 8-2, we see that critical length of ¾ bare PAX pipe in practical applications is sometimes being exceeded. For example, at an entering hot water temperature of 115 F, and a flow rate of 0.5 gpm, figure 8-2 shows a critical length of approximately 59 feet when the surrounding air temperature is 32 F. At an air temperature of 67.5 F, critical length under those same conditions is around 109 feet. Either way, lengths greater than this were often seen during the site visits. To compensate for this, water heater setpoint temperatures would need to be set higher, incurring increased standby heat loss from the tank as well as increased piping heat loss. In the case of 32 F air, figure 8-2 shows that for a 200 ft long bare ¾ PAX pipe flowing at 0.5 gpm, entering water temperature would need to be at least 145 F in order to obtain 105 F at the outlet of the pipe. Increasing water heater delivery temperature to this level increases both pipe heat loss and tank heat loss substantially.

Percent of Invested Energy Lost to Ambient

The peak UA factors of figure 8-1 can also be used to calculate the percentage of heat energy put into the water to raise its temperature that is lost by heat transfer to ambient ("Percent of Invested Energy Lost"). This "Percent of Invested Energy Lost" can be determined vs pipe length, as a function of pipe configuration, flow rate, and temperatures. When doing this calculation it is important to recognize that there is a discontinuity in the energy lost when the water in the pipe drops below a useful temperature. Once water drops below the specified minimum useful temperature (we define as 105 F), the remaining energy in the water in the pipe is no longer useful and is assumed dumped to drain or otherwise lost. "Percent of Invested Energy Lost" is a function of pipe configuration, flow rate, and entering hot and cold water and air temperature, and can be plotted in a variety of ways. Figure 8-3 shows "Percent of Invested Energy Lost" vs flow rate, cross-plotted against pipe length, for bare ¾ inch PAX piping in 67.5 F air, with entering hot and cold water temperatures of 115 F and 58 F respectively. Figure 8-3 uses peak UA values vs flow rate from figure 8-1.

At the conditions used in figure 8-3, we see that a 0.5 gpm flow rate in 100 feet of ¾ bare PAX piping would loose approximately 16 % of the total energy put into the water. At a slightly lower flow rate of 0.46 gpm, 100% of the input energy would be lost because 105 F water could not be delivered. It is this discontinuity that causes AF/PV ratios to go rapidly to infinity at low flow rates when pipe heat loss exceeds critical levels. The result in practice is that water heater setpoint temperatures are increased to compensate. Note also that the energy that needs to be provided to compensate for pipe heat loss is a function of the entering cold water temperature rather than the air temperature, once the delivered temperature drops below 105 F. The energy to heat the water drawn is also a function of entering cold water temperature. Thus as entering cold water temperature increases, the ratio of heat loss/invested energy increases because the invested energy decreases.

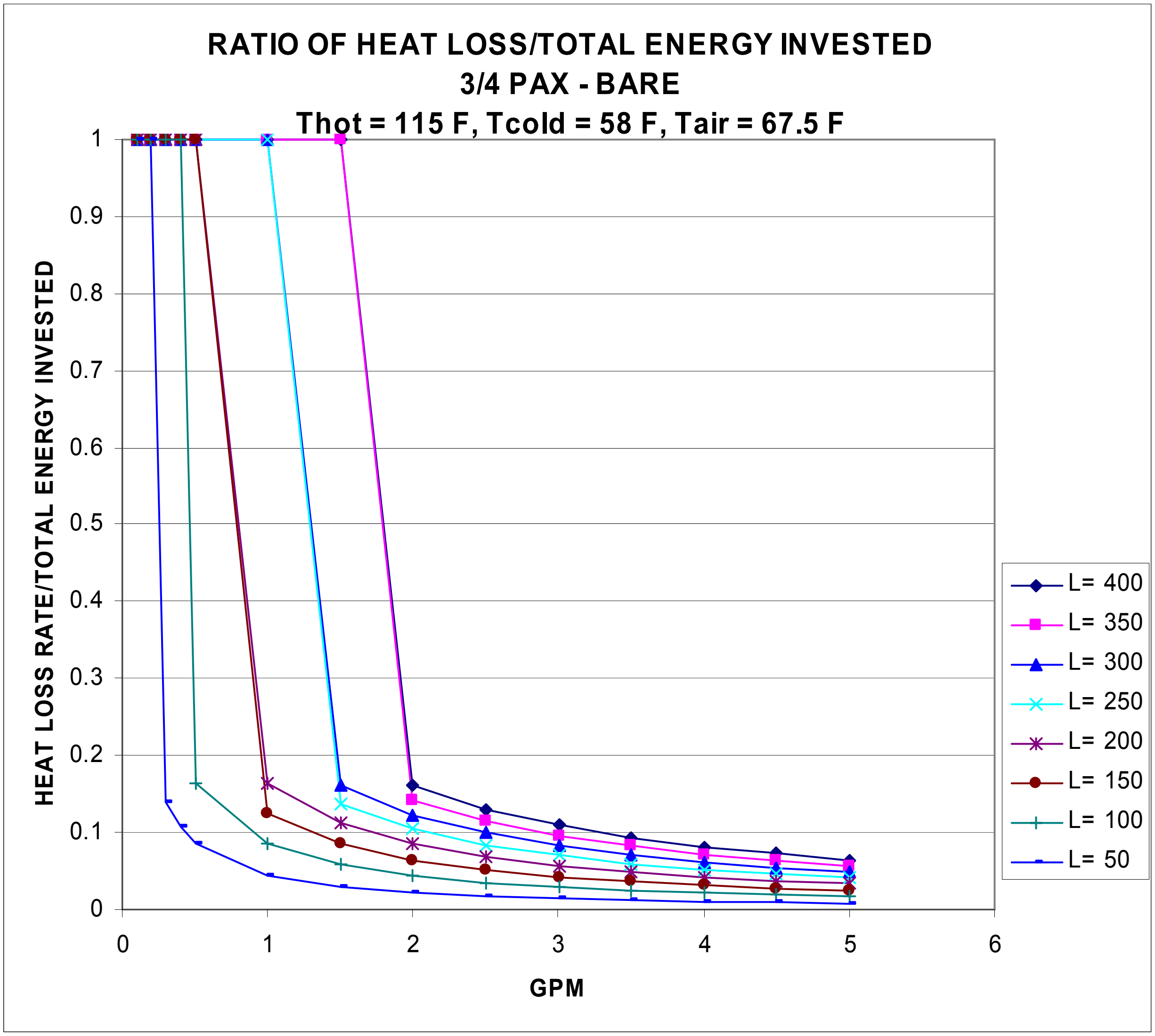


Figure 8-3
Percent of Invested Energy Lost to Ambient vs Flow Rate and Pipe Length for Bare ¾ PAX Pipe (Thot = 115 F, Tcold = 58 F, Tair = 67.5 F)(Legend is pipe length in feet.)

9

DELIVERY-PHASE AF/PV COMPARISONS – HORIZONTAL PIPES IN AIR

Comparing the AF/PV plots of sections 5, 6, and 7, we can draw some general conclusions about AF/PV ratios for the three different pipe sizes and types tested. In general, the AF/PV ratios were in similar ranges, i.e. mostly in the range of 1.1 to 1.6. However, the ¾ PAX piping had a lower bound AF/PV ratio of 1.0 under a broader range of flow conditions than did the copper pipes tested. The ¾ inch PAX piping was therefore the same, or superior to the ¾ inch rigid copper piping, depending on flow rate in the pipe and its length. Moreover, the heat loss characteristics of uninsulated ¾ inch PAX piping appeared worse than ¾ inch rigid copper (partly due to its larger outer diameter and hence larger surface area), while when insulated it had superior (lower) heat loss characteristics. Remember, however, that the smaller ½ inch diameter pipe had a smaller pipe volume per foot, which means that it has a lower wasted gallons per foot than the larger pipes under similar AF/PV ratios. In general one should use the smallest diameter piping possible from a flow rate and pressure-drop perspective when serving fixtures.

In order to perform meaningful comparisons of the delivery-phase performance of the different piping configurations tested, it will be necessary to compare them under flow rate and temperature and pipe length conditions that are likely to be encountered in real applications. Remember that AF/PV ratio is a strong function of temperature drop ratio (TDR), where TDR = $(T_{hot}$-105 F$)/(T_{hot}-T_{pipe\ initial})$. Since initial pipe temperature is a function of the pipe insulation level, environmental conditions, initial entering hot water temperature, and time between draws, TDR and AF/PV ratio are also functions of those parameters. Performing a quantitative comparison of the different piping systems under realistic conditions is beyond the scope of the current work. However, section 10 shows some HWD system example time, water, and energy waste calculations using the data presented in this report.

10
EXAMPLE CALCULATIONS USING AF/PV AND UA RESULTS

In order to demonstrate proper use of the UA and AF/PV results presented in this report, we demonstrate here the use of those results to compute the time-, water- and energy-waste of one of the piping configurations. This is done through three simple examples. All three examples assume we wish to obtain a flow rate of 1.0 gpm of water at 105 F from the fixture, for 5 minutes. We assume the same environmental conditions, end-use draws, and other parameters for all three examples. The first example assumes two draws spaced 12 hours apart. The second example assumes two draws spaced 6 minutes apart. The third example assumes two draws spaced 15 minutes apart. The assumed example conditions are:

$T_{hot\ water}$ = 115 F
$T_{cold\ water}$= 58 F
T_{air} = 67.5 F
L = 100 ft of pipe
Fixture flow rate = 1.0 gpm at 105 F for 5 minutes
Assume bare ¾ inch copper pipe in air

The water waste is determined by the AF/PV ratio and the pipe length, where the AF/PV will be selected based on the appropriate piping configuration and flow rate, and the TDR based on the temperature to which the piping cools down between draws.

Example 1 – 12 Hour Draw Spacing

First we must iterate to determine the actual flow rate of hot water in the pipe needed to deliver a mixed temperature of 105 F. In reality, this flow rate will change as delivered water temperature increases, decreasing from the full 1.0 gpm flow rate, to something less as the delivered hot water temperature increases above 105 F. In a computerized analysis, we would determine a new estimate of pipe flow rate in each time step. For the simple hand calculations to be done here, we will assume the full 1.0 gpm flow rate in the hot water pipe until "warm enough" (105 F) water reaches the fixture. After that we will assume a flow rate as calculated by a mixing energy balance, assuming fully hot water at the temperature leaving the hot water pipe is mixed with the entering cold water temperature to obtain the 1.0 gpm flow rate of 105 F water.

By an energy balance on the mixing of hot and cold water to obtain the desired mixed end-use temperature, it can be shown that the ratio of hot water flow rate to total flow rate is:

$(m_{hot}/m_{total}) = (T_{warm} – T_{cold})/(T_{hot}-T_{cold})$
$(m_{hot}/m_{total}) = (105 – 58)/(115-58)$
$(m_{hot}/m_{total}) = 0.824$

Hence our first estimate of the actual flow-rate of hot water in the pipe is:

Actual hot water flow rate first estimate = (0.824)(1.0 gpm)
Actual hot water flow rate first estimate = 0.824 gpm

Interpolating in table 8-4, we estimate that the actual temperature drop of water flowing through the pipe is 4.8 F, meaning that a second estimate of the hot water temperature reaching the fixture is 115-4.8 = 110.2 F. Using this value in the mixing expression above yields a new estimated hot water flow rate ratio of (105-58)/(110.2-58) = 0.90 and hence an new estimated hot water flow rate of 0.90 gpm. This yields a new steady-state temperature drop estimate from interpolation in table 8-4 of 4.4 F. The next estimate of delivered hot water temperature is hence 115-4.4 = 110.6 F This yields a new estimated hot water flow rate ratio of (105-58)/(110.5-58) = 0.89, and a new estimated hot water flow rate in the pipe of 0.89 gpm. Continuing to iterate we find that delivery temperature is 110.5 F and hot water flow rate is 0.895 gpm. Note that if we had assumed Thot = 135 F, the actual hot water flow rate in the pipe would have been closer to 0.68 gpm.

For the bare ¾ inch copper pipe in air, the first draw is assumed to begin from a cold-start condition where the initial pipe temperature equals the air temperature of 67.5 F. The TDR for the first draw is hence:

TDR = (115-105)/(115-67.5)
TDR = 0.21

If this were a computerized analysis we would subdivide the piping analysis into shorter length segments and estimate a new entering TDR for each pipe segment as temperature dropped in the pipe. For this simple hand-calculation we will use only the inlet TDR. From table C-6, and assuming that behavior at 100 feet is approximately the same as at the 86.3 feet of the test apparatus, we see that at 1.0 gpm flow rate AF/PV ≈ 1.395. Since, as seen in Appendix C the actual piping ID = 0.785 inches, the actual pipe volume of the 100 ft long pipe is 2.51 gallons. The wasted gallons for the first draw are then:

Wasted water = (2.51 gallons)(1.395)
Wasted water = 3.50 gallons

Assuming the hot water flow is the full 1.0 gallons per minute until the minimum 105 F hot water temperature arrives at the fixture, the time wasted waiting for "warm enough" (105 F) water to arrive at the fixture is:

t_{wait} = (3.50 gallons)/(1.0 gallons/minute)
t_{wait} = 3.50 minutes.

From table 8-2, we see the time for the pipe temperature to drop below 105 F is approximately 9.1 minutes. Using the steady-state temperature drop computation above, we can calculate the average pipe water temperature at the beginning of cool-down as:

Tpipe avg.initial = (115 + 110.5)/2
Tpipe avg.initial = 112.75 F

We can estimate the actual pipe temperature at 12 hours between draws by using the exponential temperature decay equation discussed in Appendix B. As shown in Appendix C, the mCp for bare ¾ inch rigid copper pipe is approximately 0.2084 + 0.0417 = 0.2501 Btu/ft F. And from figure 8-1 or table 8-1 we see that zero-flow UA for that pipe is 0.388 Btu/hr ft F. Thus:

(t)(UA)/(mCp) = (12 hrs)(0.388 Btu/hr ft F)/(0.2501 Btu/ft F)
(t)(UA)/(mCp) = 18.6

Then:

$T_{pipe\ 2} = T_{air} + (T_{pipe\ 1} - T_{air})e^{-18.6}$
T_{pipe} at12 hours = 67.5 + (112.75-67.5) $e^{-18.6}$
T_{pipe} at12 hours = 67.5+ (47.5)($1.0\text{x}10^{-8}$)
T_{pipe} at12 hours = 67.5 F

In other words, the pipe completely cools back down to surrounding air temperature in 12 hours. The total wasted water from the two draws is hence:

Total Wasted Water = (2)(3.50 gallons)
Total Wasted Water = 7.0 gallons

And the total wait time is:

Total Wait Time = (2)(3.50)
Total Wait Time = 7.0 minutes.

HWD system energy loss consists of four major components:

1. "Down-the-drain" energy loss caused by heating the water that displaces the warm, but not-hot-enough water wasted down the drain while waiting for "hot enough" water to arrive at fixtures.
2. Energy lost from pipe to environment during delivery and use phases of flow
3. Energy lost from pipe to environment during pipe cool-down phase
4. Energy waste due to pumping and water treatment and disposal (we will neglect this here).

Cool-down energy loss is determined using the formula

$Q_{cd} =(mCp)_{pwi}(T_{pipe\ 1} - T_{pipe\ 2})$

Where Q_{cd} = total energy loss during cool-down

$(mCp)_{pwi}$ = sum of mass times specific heat of pipe + water + insulation
$T_{pipe\ 1}$ = initial pipe temperature
$T_{pipe\ 2}$ = final pipe temperature

Calculation of Q_{cd} is straightforward when the water in the pipe has not cooled below the 105 F usable temperature. However, if the pipe has cooled below the usable 105 F, then the warm, but not hot-enough water remaining in the pipe must be wasted to drain, causing an additional rapid large energy loss during the next draw. In other words, the "cool-down" energy loss becomes a "down-the drain" energy loss. Moreover, the energy that must be supplied to make up for that extra rapid "down-the-drain" cool-down energy loss must be supplied by the water heater, heating water from cold to hot. This extra energy loss is hence not computed on the basis of heat loss to ambient, but rather on the basis of makeup water that must be heated. Note that once the pipe temperature drops to below the usable 105 F level, we don't need to use the Q_{cd} formula above at all. The "cool-down" energy loss computation in the event the pipe cools below 105 F is more simply computed directly from the AF/PV ratio of the next draw and the known pipe volume, and including the entire water waste, not just the incremental water waste above pipe volume. This alternate cool-down energy calculation is computed as follows:

$Q_{cd\ alt} = (mCp)_{water}(T_{hot}-T_{cold})$

Where:

$Q_{cd\ alt}$ = The cool-down energy loss when the pipe has cooled below 105 F
$(mCp)_{water}$ = the mass times specific heat of the total amount of water wasted to drain in the next draw after the cool-down.
T_{hot} = Water heater delivery temperature from the water heater
T_{cold} = entering cold water temperature to the water heater

Note too that the real energy cost to provide this heated water should include the tank heat input efficiency. We do not do this calculation here to avoid confusion, but it should be included in a more detailed analysis using real draws, which should also including tank standby heat loss effects..

The energy lost due to heat transfer to the environment during the delivery- and use-phases can be calculated by the equation $Q_{hl} = UA(T_{pipe\ avg.}-T_{air})$, or more correctly using log-mean temperature difference instead of $(T_{pipe\ avg.} - T_{air})$ (see an introductory heat transfer textbook for information on log-mean temperature difference). This value will also equal Q_{hl} = (water mass flow rate)(Cp)($T_{hot\ in} - T_{hot\ out}$), where Cp = 1.0 Btu/lbm F (the specific heat of water). In this example:

Q_{hl} = (0.895 gallons/minute)(5 minutes)(8.4 lb/gallon)(1.0 Btu/lb F)(115 – 110.5 F)
Q_{hl} = 6.8 Btu per draw period.

In this first example, since we have determined that the pipe cools below 105 F during the cool-down phase, all of the cool-down energy loss is lumped into the "down-the drain" energy loss, and there is no separate cool-down energy loss calculation.

The energy lost due to the "down-the-drain" water wasted is calculated as:

$Q_{cd\ alt}$ = (3.50 gallons)(8.4 lb/gallon)(1 Btu/lb F)(115 – 58 F) = 1676 Btu/draw

Total HWD system energy waste is hence:

Qtotal = (2)(1676) + (2)(6.8) = 3366 Btu

Example 2 – 6 Minute Draw Spacing

Using the exponential temperature decay equation, we see that:

(t)(UA)/(mCp) = (6 minutes)(0.388 Btu/hr ft F)/[(0.2501 Btu/ft F)(60 minutes/hr)]
(t)(UA)/(mCp) = 0.155

And hence that:

Tpipe 6 minutes = 67.5 F + (112.75-67.5)$e^{-0.155}$
Tpipe 6 minutes = 106.3 F.

We see that after the first draw the pipe does not cool down to below 105 F before the second draw occurs. This means that the water in the pipe is still at a usable temperature, and there is no incremental water waste at all in the second draw. The total water waste is hence 3.50 gallons and the total time waste is 3.50 minutes.

The energy lost to ambient during the second cool-down is the same as in the first example. However, the energy lost to ambient during the first short cool-down is much less than in the first example. The energy lost during the first cool-down is:

Q_{cd} = (0.2501 Btu/ft F)(100 ft)(112.75 – 106.3 F)
Q_{cd} = 161.3 Btu.

Total delivery system energy waste is hence:

Qtotal = (1)(1676) + (1)(161.3) + (2)(6.8) = 1851 Btu.

Example 3 – 15 Minute Draw Spacing

Calculations here are similar to examples 1 and 2 above, except that after the first draw, the pipe cools down below a usable temperature, but not all the way back down to ambient temperature. This means we must determine a different TDR and AF/PV ratio for the second draw than for the first.

Using the exponential temperature decay equation discussed in Appendix B, we see that:

(t)(UA)/(mCp) = (15 minutes)(0.388 Btu/hr ft F)/[(0.2501 Btu/ft F)(60 minutes/hr)]
(t)(UA)/(mCp) = 0.388

And hence that:

Tpipe 15 minutes = 67.5 F + (112.75-67.5)$e^{-0.388}$
Tpipe 15 minutes = 98.2F.

The energy lost during the second cool-down is the same as for example 1. The energy lost during the first cool-down is computed by the same method as for example 1, except that the AF/PV ratio for the second draw will be different. The TDR for the second draw is:

TDR = (115-105)/(115-98.2)
TDR = 0.595

From table C-6, and assuming that behavior at 100 feet is approximately the same as at the 86.3 feet of the test apparatus, we see that at 1.0 gpm flow rate AF/PV ≈ 1.20. Since, as seen in Appendix C the actual piping ID = .785 inches, the actual pipe volume of the 100 ft long pipe is 2.51 gallons. The wasted gallons for the second draw is then:

Wasted water 2^{nd} draw = (2.51 gallons)(1.20)
Wasted water 2^{nd} draw = 3.01 gallons

The total water wasted in this example is hence:

Total Wasted Water = 3.50 + 3.01 gallons
Total Wasted Water = 6.51 gallons

Similarly, since the flow rate is 1.0 gpm, the total wasted time is 6.51 minutes.

The wasted energy due to the second cool-down is the same as in example 1. The cool-down wasted energy waiting for "warm enough" water to arrive at the fixture in the second draw is:

$Q_{cd\ alt\ 2nd\ draw}$ = (3.01 gallons)(8.4 lb/gallon)(1 Btu/lb F)(115 – 58 F)

$Q_{cd\ alt\ 2nd\ draw}$ = 1441 Btu

Total HWD system energy waste is hence:

Qtotal = (1)(1676) + (1)(1441) + (2)(6.8) = 3131 Btu

Note the large difference in energy waste from the piping system due to a few minutes extra delay in taking the second draw. Pipe insulation helps considerably in extending the allowable delay.

Importance of Tank Heat Loss

The above three simple examples all compared the same length piping. When doing real evaluations of alternate HWD system piping configurations, the piping lengths will be different from system to system, because the water heater(s) may be in different places in the structure. Water flowing in shorter pipes has less temperature drop than in longer pipes. Therefore, to obtain the same temperature water reaching the fixture, the delivery temperature from the water heater will need to be higher for the longer pipe runs compared to the shorter. The increase in delivery temperature necessary is usually in the 5 to 10 F range, and is sometimes as large as 40 - 50 F. An increase of even 5 F in tank delivery temperature can significantly increase yearly energy use, and must be included in the overall HWD system energy waste computation. This is because the heat loss from the tank increases as its delivery temperature increases. Note that the increased tank heat loss happens 24 hours per day, 365 days per year, and is not a function of number of draws or their time spacing, which is why the effect is significant. Moreover, the tank heat loss is made up by the heat input device for the tank, whose heat input efficiency must also be considered.

Recirculation System Energy Loss

It should be noted that if a recirculation-loop system is used, much (but not all) of the "down-the-drain" water waste is eliminated. However, much of the "down-the-drain" energy waste still remains, but in a different form. In the RL case, instead of energy being wasted heating make-up water wasted down the drain, energy is lost from the return line to the tank, which did not exist in the non-RL systems. Depending on how the RL system is designed and operated, this energy loss may be much larger, or it may be smaller than the down-the-drain energy waste. Investigation of RL system attributes was beyond the scope of the current work.

Note, however, that the UA factors developed in the current work can be directly applied to calculating heat loss off of the RL piping. Most of the energy-loss attributes of RL systems can hence be calculated using the data from this report. The difficulty in calculating RL system energy loss is in knowing how the RL system controls are supposed to function, vs how they really function in the field. Water waste occurs in RL systems both because of the need to purge branch-lines of cold water during draws, and because some RL system controls prevent the system from functioning as an RL system under some conditions, causing it to revert to full down-the-drain water waste behavior.

11
TEST RESULTS – HORIZONTAL ¾ INCH RIGID COPPER PIPE IN WATER AND SIMULATED WET AND DAMP SOIL ENVIRONMENTS

The site visits described in section 3 revealed that the under-slab buried-pipe environment was always at least damp, and was sometimes fully-saturated to the point of having standing liquid water. Several laboratory tests were performed to gain some quantitative, but mostly qualitative understanding of the impact these wet or damp environments would have on HWD piping performance. The tests performed were not meant to be definitive regarding pipe performance in buried environments. Rather, brief tests were performed to understand and quantitatively bound the worst-case scenario (fully-submerged bare pipe in liquid water) and to approximate some other less-wet conditions. These tests were mostly done to gain an understanding of test equipment and fixture requirements for future buried-pipe tests. Extensive test sequences were not performed for the wet and damp environment tests as they were for the in-air tests.

Several different tests were done on a section of horizontal ¾ inch rigid copper pipe in wet and damp environments.

- Piping submerged in liquid water
- Piping wrapped in dry-towels (baseline for wet tests)
- Piping wrapped in towels submerged in liquid water (simulated porous wet soil tests)
- Piping wrapped in damp towels (wetted then suspended in air until dripping stopped – simulated porous damp soil tests)

Tests were done using water and towels instead of real soils to facilitate rapid testing. It was easy to flush and refill the water bath between tests, enabling multiple tests to be performed in a single day. Use of real soils would likely have limited testing to one test per day.

Water Bath Test Fixture

Figures 11-1 and 11-2 show photographs of the water-bath test fixture. Preliminary analysis had revealed that piping heat loss would be so much higher in wet and damp environments compared to the in-air environment, that the water bath should not be applied to the entire test section. Rather, as shown in Figures 11-1 and 11-2, the wet and damp tests were performed on only the first half of the ¾ inch rigid copper pipe test section. The same copper test section was used as was used for the in-air tests. The water bath fixture consisted of two 20 ft long sections of 7 inch deep galvanized eave-trough (gutter), with a connecting u-bend. The entire bath fixture was lined with a continuous plastic liner to ensure against water leaks. The plastic was secured to the pipe ends near the water heater tank. Total water bath length was approximately 43 feet. As seen in figures 11-1 and 11-2, the bare copper pipe was spaced away from the sides and bottom of the gutter using short sections of 1 inch nominal diameter PVC piping.

Water Bath Test Results – Bare Pipe In Liquid Water

Water depth for these tests was approximately 5-6 inches, with the copper pipe approximately 1 inch from the bottom and sides. The water bath was drained and refilled for each test, to ensure uniform low water temperature. It was not possible to separately vary initial pipe water temperature and water bath temperature because the high heat transfer rates observed caused them to rapidly reach the same temperature. Likewise, air temperature was relatively unimportant during these tests since the pipe was in contact with the water bath instead of air. Tests were performed only with entering hot water temperatures in the 133-137 F range, and initial water bath temperatures in the 60 – 90 F range. Note that unlike in-air tests, as the tests proceeded, an axial temperature gradient developed in the water bath. This was taken into account when calculating UA values.

Figure 11-3 shows both the scatter-plot test results and curve-fits to the measured pipe heat loss UA values for the bare ¾ inch rigid copper pipe in the water bath, and also for the wet-towel and damp-towel tests (see discussions in following paragraphs). The curve fit UA plot for the bare in-air case is also shown in figure 11-3. On the scale necessary to show the water-bath UA values, the bare in-air plot is just barely visible on the bottom, having a peak value of 0.44 Btu/hr ft F. We see that the UA values of the ¾ rigid copper pipe in the water bath ranged from 34 to 75 times higher than for the bare pipe in air.

Figure 11-1
Water Bath Test Fixture

Figure 11-2
Bare ¾ Inch Copper Pipe In Water Bath Test Fixture

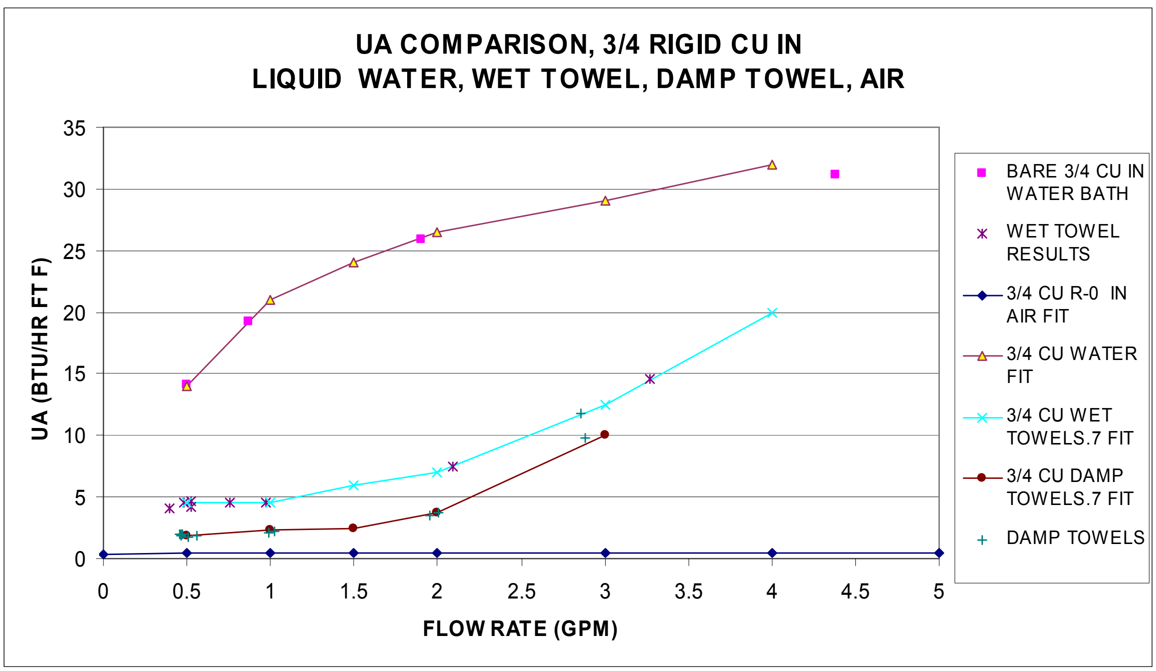


Figure 11-3
UA Values of ¾ Inch Rigid Copper Pipe in Air vs Wet and Damp Environments

Test results showed that heat loss from the pipe to the water was so high that at low flow rates water at 105 F could not be delivered until the water bath itself heated up. AF/PV ratios were infinite until flow rates of around 2 gpm, even at the high TDR values tested (TDRs tested were in the range of 0.413 – 0.437). Even at flow rates above 2 gpm, AF/PV ratios were high compared to in-air values at similar TDRs. For example at TDR = 0.426 and 4.4 gpm flow rate, AF/PV ≈ 1.49 at the 43 foot location, whereas a similar TDR and flow rate in air yielded AF/PV ≈ 1.2 Moreover, at the higher flow rates, which could at least deliver 105 F water, temperatures in the pipe dropped to below usable within one minute of flow stoppage. This was because the water bath was still at a much lower temperature than the water in the pipe. Figure 11-4 compares the AF/PV ratio plots for the 43 foot pipe length, for bare ¾ rigid copper pipe in air vs in liquid water, vs in the wet-towel tests and the damp-towel tests. The lowest plot in figure 11-4 is the in-air result, the next highest is the damp-towel result, followed by the wet-towel result, and finally the liquid water-bath result.

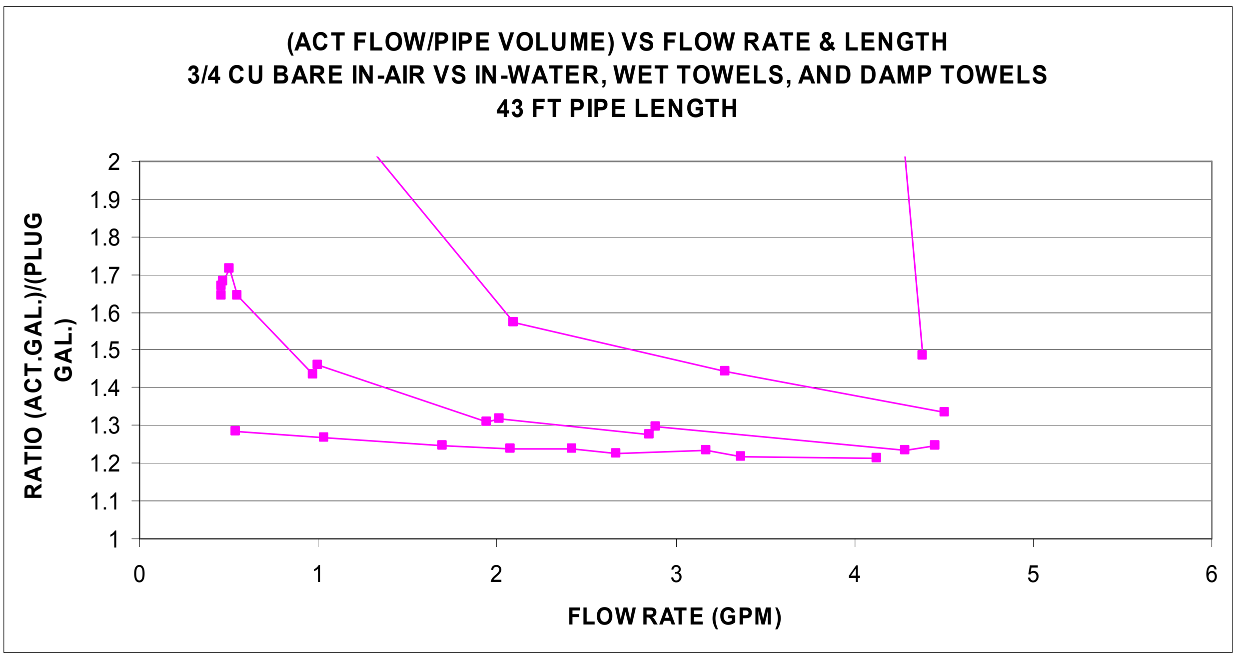


Figure 11-4
AF/PV Ratio Plots for ¾ Inch Rigid Copper Pipe in Air vs Wet and Damp Environments (Lowest Plot is in-air, second up is damp towels, third up is wet towels, highest is in liquid water TDR = 0.41 to 0.47)

Dry Towel Tests In Air

In preparation for wet- and damp- simulated soil tests, the pipe was wrapped tightly in thick towels and tested laid in the dry trough. Resultant towel radial thickness was around 1.0 to 1.5 inches. Small-diameter rope was spiral-wound around the towels to hold them tightly to the pipe. Several spot-tests were done to develop baseline performance data for this configuration compared to the wet- and damp-towel tests to be performed later (see below). As expected, the heat loss characteristics of the dry-towel configuration were better (lower) than for bare pipe, but not as good as foam pipe insulation. The AF/PV ratios observed were similar to the bare in-air tests (which were similar to the insulated tests) except for elimination of the region A behavior discussed in Figures 4-2 and 4-3, which is impacted by heat loss to the surroundings.

Simulated Wet Porous Soil Tests – Wet Towels In Liquid Water

In order to approximate performance of pipe buried in porous wet soil, the pipe with towels tightly bound to it (see Dry Towel Tests In Air above) was laid in the test water-bath trough and completely submerged in water. As seen in figure 11-5, water level was essentially to the top of the towels. Test procedures were as described above for the bare-pipe liquid water bath tests.

As seen in figure 11-3, measured UA values of the simulated wet-porous-soil tests (wet towel tests) were much higher than for bare pipe in-air tests, but much lower than for bare-pipe in liquid water. Measured UAs for this configuration were from around 4.6 Btu/hr ft F for pipe water flow rates in the 0.5 to 1.5 gpm range, up to 14.6 at 3.3 gpm. The measured UA values ranged from 10 to 40 times higher than for the bare-pipe in air, but only 1/3 to 1/2 of the value for bare pipe in liquid water. While this test does not necessarily represent the behavior of pipe buried in real soil, it does suggest that the presence of liquid water in loosely packed soil can increase pipe heat loss rate substantially compared to cases where the water is not present. An unknown yet to be examined is the possible development of prolonged heat energy storage in the soil and water around the pipe, which could alter performance after repeated draws over an extended period.

Figure 11-4 compares the AF/PV ratio plots for the 43 foot location of the bare-pipe in-air tests to the wet- and damp-towel test and liquid water-bath results for that location and similar TDRs (TDR = 0.41 – 0.47 range). The wet-towel result is the third plot from the bottom in figure 11-4. AF/PV ratios were significantly higher for the wet-towel tests than for bare pipe in air. Only at the highest flow rates and high TDRs did the wet-towel AF/PV ratios approach the in-air values. Moreover, the shapes of the AF/PV curves for the wet-towel tests clearly showed that all were in the region where heat transfer to the environment was a significant effect – that is, all had the characteristic shapes of region A shown on figures 4-2 and 4-3. Note again, however, that these results do not necessarily represent the behavior of pipe buried in real soil. Rather, they merely suggest that the presence of liquid water in loosely packed soil can increase pipe heat loss enough to significantly increase AF/PV ratios. Likewise, an unknown yet to be examined is the possible development of prolonged heat energy storage in the soil and water around the pipe, which could alter performance after repeated draws over an extended period.

Figure 11-5
Simulated Saturated Porous Soil Test (Wet Towel Test) – Bare ¾ Inch Copper Pipe

Simulated Damp Porous Soil Tests - Damp Towels

In order to approximate performance of pipe buried in porous damp soil, the pipe with towels tightly bound to it (see dry-towel and wet-towel discussions above) was suspended above the water-bath trough. The suspended towel-wrapped pipe was then thoroughly wetted and allowed to remain until liquid water stopped dripping from the towels. Once dripping had stopped, tests were run as in the in-air and water-bath tests described above. It was noted during testing that due to the weight of the wetted towels, the towels pulled away from the bottom half of the pipe in this suspended configuration. This probably lessened the effect the damp towels had on the pipe and made performance under this configuration more like the in-air tests than it would be in a truly buried environment. Note that in this configuration, evapo-transpiration, where water evaporates and then travels through the porous medium, is an important heat transfer mechanism.

As seen in figure 11-3, measured UA values of the simulated damp-porous-soil tests (damp towel tests) were much higher than for in-air tests, but much lower than for bare-pipe in liquid water and wet-towel tests. Measured UAs for this configuration were from around 1.9 Btu/hr ft F for pipe water flow rates in the 0.5 gpm range, up to 10.6 at 3 gpm. The measured UA values ranged from 4.6 to 20 times higher than for the bare-pipe in air, but only 1/6 to 1/4 of the value for bare pipe in liquid water. While this test does not necessarily represent the behavior of pipe buried in real soil, it does suggest that the presence of moisture in loosely packed soil can increase pipe heat loss rate substantially compared to cases where the water is not present. An unknown yet to be examined is the possible development of prolonged heat energy storage in the soil and water around the pipe, which could alter performance after repeated draws over an extended period.

Figure 11-4 compares the AF/PV ratio plots for the 43 foot location of the bare-pipe in-air tests to the wet- and damp-towel test and liquid water-bath results for that location and similar TDRs (TDR = 0.41 – 0.47 range). The damp-towel result is the second plot from the bottom in figure 11-4. AF/PV ratios were significantly higher for the damp-towel tests than for bare pipe in air. Only at the higher flow rates and high TDRs did the damp-towel AF/PV ratios approach the in-air values. Moreover, the shapes of the AF/PV curves for the damp-towel tests clearly showed that all were in the region where heat transfer to the environment was a significant effect – that is, all had the characteristic shapes of region A shown on figures 4-2 and 4-3. The trends shown in figure 11-4 indicate that at flow rates below about 2 gpm, the damp-towel environment has such high heat transfer that it significantly affects the AF/PV ratio, even at high TDRs. At flow rates above 2 gpm, heat transfer to ambient is still important in the damp-towel tests, but that heat transfer has a less significant impact on AF/PV ratio. Most flow rates of interest in residential applications are below 2 gpm. Note again, however, that these results do not necessarily represent the behavior of pipe buried in real soil. Rather, they merely suggest that the presence of moisture in loosely packed soil can increase pipe heat loss enough to significantly increase AF/PV ratios. Likewise, an unknown yet to be examined is the possible development of prolonged heat energy storage in the soil and water around the pipe, which could alter performance after repeated draws over an extended period.

12
TEST RESULTS – VERTICAL ¾ INCH RIGID COPPER PIPE IN AIR

Vertical Pipe Test Fixture and Procedures

Tests were performed on one vertical pipe test section. As noted in Appendix B, the vertical test section consisted of one continuous 20 foot ¾ inch rigid copper pipe section, with a U-bend at the upper end of the section similar to those used in the horizontal ¾ inch rigid copper pipe tests. Figures B-5 and B-6 show the vertical piping configuration. The vertical test section was connected to the outlet of the horizontal ¾ inch rigid copper test section, which was insulated with ¾ inch thick R-4.7 foam pipe insulation. Various types of heat isolation couplings and unions were tried between the horizontal and vertical pipes. A full-port ball-valve was installed at the lower entrance to the vertical test section so that the vertical pipe section could be kept isolated from the horizontal piping while the latter was being primed with hot water prior to vertical pipe tests. Immersion thermocouples were installed at the inlet and outlet of the vertical pipe test section, and external clamp-on thermocouples were installed at several locations along the pipe. A flow-adjust valve and a ball-valve were installed at the outlet of the vertical test section U-bend.. A hose was connected to the outlet of the vertical test section U-bend to bring water back down to ground level for disposal.

A typical draw test on the vertical test section consisted of first adjusting the test flow rate with the flow adjusting valve at the exit of the vertical test section. Initial pipe temperature conditions were also established at this time. Next the horizontal pipe was primed with hot water until its outlet temperature remained constant for several minutes. The lower ball-valve on the vertical test section was then opened, followed as closely as possible by the upper (outlet) ball-valve. Time between opening of the lower ball-valve and initiation of flow by opening the upper ball-valve was typically 10-12 seconds. This was a longer period than for the horizontal pipe tests because test personnel had to rapidly climb the ladder leading to the outlet of the vertical test section. Both the ladder and the vertical test section were suspended from the ceiling. Both insulated and uninsulated vertical pipe tests were performed.

Vertical Pipe Test Results

UA values appeared similar to those seen in the horizontal piping, so results will not be repeated here. There may have been some differences, but they were not beyond the normal variation seen due to experimental accuracy, since results were available from only approximately 21 feet of pipe.

The AF/PF ratio behavior was also generally similar to the horizontal cases of similar length except that it did not appear that the stratification effect (Region B in Figures 4-2 and 4-3) developed in vertical pipe. Rather, the vertical natural-convection circulation at the hot/cold interface just behaved as if it were normal flow-induced mixing. The natural convection mixing did not produce an extended hot/cold interface and hence became self-limiting fairly quickly. The vertical test section AF/PV plots are therefore not repeated here –they are like the horizontal tests except the higher AF/PV ratios due to region B do not exist. However, the behavior of the vertical test section with respect to buoyancy-driven effects was not what was expected, as explained in the next paragraphs.

Some interesting unexpected behavior was observed with respect to the impact of convective heat loss and interaction between the horizontal and vertical section. A number of tests were performed where the valve at the base of the vertical test section was opened during flow from the horizontal section or during cool-down of the horizontal section. This opened the cold vertical pipe to communication with the hot horizontal pipe but did not initiate flow in the vertical pipe, which was controlled by a valve at its outlet at the top. It was anticipated that natural convection and buoyancy effects would cause warm water to rise to the top of the vertical test section – but this did not occur. Instead the temperature of the lower portion of the vertical pipe increased a small amount, but then reached a steady-state value. Similarly, the temperature of the horizontal pipe initially dropped upon opening the vertical valve but then reached a steady-state cool-down rate similar to the other sections of the horizontal pipe. Figure 12-1 shows temperature vs time plots of one such case where the entry valve to the vertical test section was opened part way into the cool-down phase of the horizontal pipe, as evidenced by the change in slope of the T5 temperature trace. In figure 12-1, T1 through T5 are temperatures from inlet to outlet on the ¾ inch copper horizontal test section, T13 is the temperature at the base of the vertical test section, and T25 is the temperature at the top. T22 through T24 are ambient air temperatures. T5 (horiz. pipe temp. at outlet) is the upper plot that changes slope at 02:50. T13 (lower part of vert. pipe temp.) is the lower plot that changes slope at 02:50.

It is believed that what happens is that heat transfer from the vertical pipe is great enough to remove the heat energy that enters the vertical test section by natural convection of the warm water. This heat removal appears to happen within the lower 3-6 feet of the vertical pipe. This is probably because the copper pipe conducts away the heat energy and spreads it throughout the surface area of the entire pipe. The result is a stable inverted temperature gradient in the vertical pipe, with warmer water remaining in the lower section of the pipe. This effect stops the natural convection flow, thus also preventing the horizontal section from dropping much in temperature. The effect happens whether or not the pipe is insulated. A slightly longer portion of the bottom of the vertical pipe warms up if it is insulated than if it is not, but still typically less than 6 feet.

It would be interesting to study this effect further, for it has implications about tank heat loss into piping and about heat loss into unused side branches from main trunk lines. Time did not permit further study, since testing other horizontal pipe materials and configurations was deemed a higher priority.

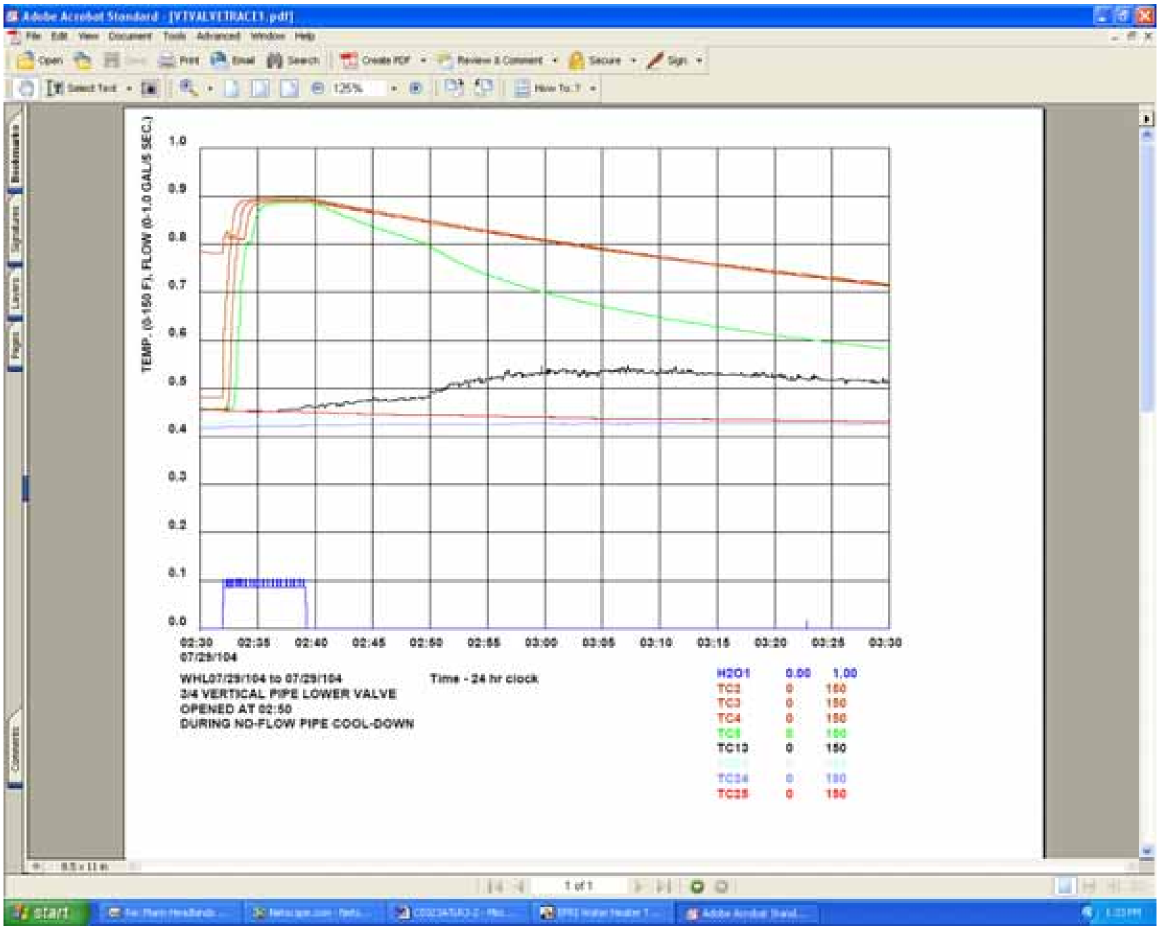


Figure 12-1
Vertical Pipe/Horizontal Pipe Natural Convection Effects on Pipe Temperature (Isolation valve opened at 02:50 – No Flow)

13 CONCLUSIONS AND RECOMMENDATIONS

Conclusions

- Analysis of site visit data shows that at least for the sites visited, length and volume of under-slab piping was greater than expected, and greater than would be typical of above-ground piping. A method of estimating piping lengths and volumes for "conventionally" laid-out HWD systems is provided in this report. "Conventional" in this case means piping that runs parallel and perpendicular to foundation walls and other structural members, as opposed to diagonally. This estimation method can be useful for comparing alternative HWD system piping layouts.

- It is fair to say that at all the sites visited, the HWD system piping layouts could have been better, which would have resulted in less length and volume of HWD piping, even if the water heater and fixtures remain in the same positions. One method of achieving this would be to use more diagonal piping runs, both in under-slab and above-ground HWD systems. Another improvement would be to pay closer attention to where pipe branches separate and to which fixtures are on which branches, such that piping runs are shorter, and use of one fixture can better provide preheating of water that would be used at other fixtures at later times. One method of improving under-slab piping lengths and volumes in particular would be to make more of the piping above-ground, thus eliminating some of the extra "up-and-down" piping length at the ends of under-slab piping where it must rise above the slab to make connections.

- For the manifold distribution system observed, piping lengths could have been dramatically shortened (many cut almost in half) if the piping had been run in the floor between levels instead of running it all to the attic and back down. Similarly, piping lengths could have been significantly reduced for some fixtures that were near other fixtures (e.g. dual sinks in bathrooms), by feeding both fixtures from a common hot water line instead of serving each with a dedicated line.

- It appeared that the under-slab environment is always at least damp, and sometimes wet to the point of having standing water in pipe trenches. Whether or not the under-slab environment stays damp over time is unknown. Tests performed here indicate, at least qualitatively, that moisture presence in the under-slab environment can substantially increase heat loss from uninsulated hot water distribution system pipe in that environment. It is unknown whether or not the under-slab environment will over time become a thermal storage medium that may lessen heat loss from buried pipes after prolonged use. Performance of insulated pipe in the under-slab environment has not yet been investigated.

- Regarding actual pipe insulation practices observed at sites visited, insulation treatment at elbows and Ts could have been much better. Additionally, butt-joints of insulation segments would perform better if glued together as per manufacturer recommendations. Moreover, insulation segments should be axially compressed during insulation to compensate for insulation shrinkage over time, as per manufacturer recommendations. These practices were not seen at any of the sites visited.

- Laboratory testing and analysis showed that the extent to which pipe insulation impacts HWD system energy waste depends on the environment where the pipes are run and the nature of the draws, especially regarding time-spacing of draws. Pipe insulation provides dramatic improvements in pipe cool-down rates, extending cool-down times by 200-400% compared to bare pipe. Slowing pipe cool-down can save significant HWD system energy loss by significantly reducing the need to purge piping of luke-warm or cold water before draws. With pipe insulation, the water in the pipe will remain above a useful temperature between many draws in real situations. Adding pipe insulation on pipes in recirculation-loop HWD systems saves significant amounts of energy if run-time of the RL system pump(s) is at all significant (i.e. greater than one hour, possibly less). The pipe heat loss (UA) factors reported here allow estimation of piping heat loss rates, pipe temperature vs time during cool-down, and more.

- Laboratory testing also showed that ½ inch thick foam pipe insulation performs almost as well as ¾ inch thick foam pipe insulation, providing heat loss rates that were within 6% of those of the ¾ inch thick insulation. The thicker insulation provided slightly greater improvements in cool-down time, ranging from 7-13% longer than for ½ inch thick foam. This difference in cool-down time could be important in real installations, depending on draw time-spacing.

- Time, water, and energy waste during delivery-phase flow have been quantified as a function of important variables in this report. The important variables include hot water temperature, initial pipe temperature, ambient temperature, pipe size, pipe length, flow rate, insulation level, and pipe material.

- It was discovered that flow during the delivery phase of a hot water draw is a highly transient, non-steady-stage phenomenon. Flow during the delivery phase (and in some cases continuing for prolonged periods of draw thereafter) was discovered to be characterized by at least 3 distinctly different flow regimes. We have termed these flow regimes stratified flow, normal flow, and slip-flow.

- At low flow rates and low initial pipe temperatures, where flow of the cold water in the HWD pipe during the draw was laminar, flow stratification was observed to occur, where hot water would flow a greater distance down the top side of the pipe than on the bottom. Flow stratification increased water and energy waste by allowing more heat transfer to occur between hot and cold water flowing in the pipe than if the stratification had not occurred.

This was especially true in pipe entrance regions. Axial heat conduction in the pipe wall also contributed to this detrimental hot-to-cold water heat transfer at low flow rates.

- At intermediate flow rates, hot water temperatures, and initial pipe temperatures, water and energy waste during the delivery phase were primarily influenced by mixing of hot and cold water in the pipe caused by flow turbulence, and by heat transfer from the hot water to heat up the mass of the pipe wall. This was the "normally occurring" mode of delivery-phase flow.

- At high flow rates and warm initial pipe temperatures, slip-flow tended to occur in the pipes. Slip-flow was characterized by shearing of the boundary layer, reducing the turbulent eddies that caused mixing in the pipe. Slip-flow reduced water and energy waste by reducing both heat transfer to ambient, and by reducing mixing of hot and cold water in the pipe. Slip flow was observed to occur more readily at lower flow rates and temperatures in some types of piping compared to others. More study of piping system configurations and materials is needed to better understand the slip-flow phenomenon and how to capitalize on it.

- Heat loss to ambient was found to significantly affect water and energy waste under low flow rates for fairly long lengths of uninsulated pipe. However, these pipe lengths and flow rates are in the range of what is commonly seen in real HWD system installations so this is a significant effect. Moreover, this effect becomes highly significant even at higher flow rates and shorter pipe lengths if the piping is in a high heat-loss environment, such as in very cold surroundings, moving air, or wet conditions.

- Time, water, and energy losses associated with vertical piping appeared similar to those for horizontal within the narrow range of vertical pipe configurations tested. The most notable difference was lack of a stratified flow regime. In vertical pipe flow stratification is replaced by full-diameter mixing, similar to "normal" flow mixing. Some unexpected behavior was observed regarding vertical natural convection flow in vertical pipes. Stable inverted temperature gradients were observed to occur in vertical pipe exposed to hot water at the bottom. It appeared that heat transfer in the pipe wall could cool the warm water enough to stall vertical natural convection flow. More investigation of this phenomenon is recommended.

- Examples are provided showing how to use the pipe heat loss and delivery-phase time and water waste findings to calculate HWD system energy losses. Tank standby heat loss is important in these energy waste calculations because some HWD system piping configurations require higher tank setpoint temperatures than others in order to deliver usable hot water at fixtures. Since tank heat loss occurs 24 hours per day 365 days per year, even small increases in tank setpoint temperature can have important energy impacts.

Recommendations

While extensive testing was performed under the current effort, it represents just a small portion of the research that is needed to fully understand performance of HWD systems. Table 13-1 shows the matrix of piping tests that have been performed to date, compared to the tests that should ultimately be performed. Note that the vertical pipe and water-bath tests that were performed are not shown in this matrix.

In addition to the need for tests on a wider variety of pipe sizes and types, there is also a need for more tests on recirculation-loop systems. These RL system tests are needed to evaluate how current system design and control strategies really work, especially from an energy waste perspective, and to identify improvements or alternatives. RL system tests are particularly important because retrofit applications, where the cold water line is used as a return-line for the RL system, are a common method of solving hot water delivery delay problems.

Table 13-1
HWD System Pipe Testing Recommended

	IN-AIR TESTS			BURIED TESTS SAND, SOIL, GRAVEL, CEMENT DRY, DAMP, WET		
PIPE	**BARE**	**½ FOAM**	**¾ FOAM**	**BARE**	**½ FOAM**	**¾ FOAM**
3/8 Cu Rolled						
½ Cu Rolled						
¾ Cu Rolled						
1 Cu Rolled						
½ Cu Rigid	DONE	DONE	DONE			
3/4 Cu Rigid	DONE	DONE	DONE			
1 Cu Rigid						
1½ Cu Rigid						
2 Cu Rigid						
½ PAX						
¾ PAX	DONE		DONE			
1 PAX						
1½ PAX						
2 PAX						
3/8 PEX						
½ PEX						
¾ PEX						
1 PEX						
½ CPVC						
¾ CPVC						
1 CPVC						
1½ CPVC						
2 CPVC						

Specific recommendations are as follows:

- More tests should be performed on more piping configurations as shown in table 13-1.
- Tests on buried pipe are especially important.
- Tests aimed at quantifying the actual energy-performance of various types of RL systems, and on developing methods to improve their performance, should be performed.
- More study of natural-convection flow in vertical pipes should be performed, since this has importance to HWD system energy losses from unused vertical branches, and from water heater tanks into piping.
- Efforts should be undertaken to develop or update user-friendly HWD system computer models so results can more readily be used for system design and analysis procedures.
- Analyses should be performed comparing the time, water, and energy waste characteristics of different HWD system designs in different applications and climates. This work should be done both as an aid to updating building energy efficiency standards, and to enable development of HWD system design manual(s).
- HWD design manuals and other technology-transfer materials and training programs should be developed to ensure the information produced reaches practitioners.

14
REFERENCES

Rohsenow & Choi, Heat Mass and Momentum Transfer, Prentice Hall, Inc., Englewood cliffs, N.J., 1961

A

APPENDIX – EXAMPLE PIPING SYSTEM LENGTH AND VOLUME COMPARISONS

In order to understand and compare the lengths and volumes of pipe in different HWD piping systems, it is instructive to compare them on a few example buildings. In this appendix we compare an above-ground trunk-and-branch system vs an under-slab piping system on a theoretical single story house that is similar to one of the site visit houses. We also compare an above-ground trunk-and-branch system vs an under/above ground piping system and vs a manifold system on an example 2-story house. In the 2-story example, the house used is approximately like the real site-visit house where the manifold system pipe length measurements were taken. Both examples are for large houses, with fixtures spread far apart, as is typical of large houses.

Example Single Story House

The approximate floor plan for the example 1-story house, showing hot-water-using fixtures and a length reference, is shown in figure A-1. This structure is similar to one of the construction sites visited. The structure has 4 bedrooms and 3.5 bathrooms. The water heater is assumed located in the garage in this example. We assume both above-and under-slab piping systems are PAX.

The above-ground trunk-and-branch system is assumed to run approximately as shown in figure A-2, with all piping in the attic except vertical in-wall drops to fixtures. We assume the main trunk line is 1 inch until after the branch to bathroom 2, where it drops to ¾ inch. We also assume the branches to bathrooms 1 and 2 are ¾ inch. We assume branches to the laundry room and the half-bath are ½ inch, and that pipes in the other bathrooms drop to ½ inch after the first fixture encountered there. We assume the branch to the kitchen is ¾ inch.

The under-slab piping system is assumed to have four branch lines, separating near the base of the water heater. One serves the kitchen, another serves bathroom 1, the third serves the laundry room and ½ bath, and the fourth serves the master bath and bathroom 2. All the under-slab main branch lines are assumed ¾ inch, traveling in series to all fixtures on the branch.

Table A-1 compares the total pipe volumes from the water heater to each fixture of Figure A-1 for the two piping schemes, including volume for vertical in-wall piping and vertical piping from the water heater up to the attic (trunk and branch) or down to the slab (under-slab). As can be seen from table A-1, under-slab piping has considerably more volume between the water heater and the fixture (and hence will have more time, water and energy waste) than the above-ground trunk and branch system, with a few exceptions. One exception is when there is a free-standing or peninsula fixture that is not directly next to a wall (e.g. the island kitchen sink in this example). In this case, the extra piping length that would be required to bring piping from the nearest wall to the island sink causes the trunk-and-branch system to have slightly greater volume than the under-slab system. The other exception is for fixtures near to the water heater (WH), where the extra volume of the longer large-diameter main trunk line increases the volume to those close fixtures slightly over what would exist in the under-slab system. Table A-1 shows the average ratio of under-slab piping system "WH –to-fixture volume" divided by above-slab trunk and branch piping system "WH-to-fixture volume" to be around 1.4.

Table A-1
Single Story House Example Pipe Volume Comparisons – Above Ground Trunk & Branch vs Under-Slab

FIXTURE	ABOVE-GROUND TRUNK & BRANCH PIPING VOLUME WH-FIXT. (gallons)	UNDER-SLAB PIPING VOLUME WH-FIXT. (gallons)	RATIO UNDER-SLAB/ TRUNK & BRANCH
Wash.Mach.&Basin	0.63	0.95	1.52
1/2 Bath Sink	0.92	1.16	1.26
Bath 2 Sink	1.44	1.34	0.93
Bath 2 Tub/shower	1.53	2.35	1.54
M.Bath Shower	2.61	4.08	1.56
M. Bath Sink	2.52	4.53	1.8
M. Bath Sink	2.47	5.00	2.02
M. Bath Tub	2.36	5.20	2.2
Kitchen Sink	4.05	3.44	0.85
Bath 1 Sink	2.67	2.85	1.07
Bath 1 Tub/shower	2.84	3.05	1.07

Example 2-Story House

The approximate floor plan for the example 2-story house, showing hot-water-using fixtures and dimension references, is shown in figures A-3 (first floor) and A-4 (second floor). This structure is similar to the site-visit house having the manifold HWD system. The structure has 4 bedrooms and 4 bathrooms. The water heater is assumed located in the garage in this example. We assume both above-and under-slab piping systems are PAX, and we use the actual pipe size and length measurements from the site for the manifold system.

The above-ground trunk-and-branch system is assumed to run approximately as shown in figures A-5 and A-6 (noting that much of the piping shown on the first-floor schematic is the same piping as shown for the second floor, since the pipe runs in the ceiling), with all piping between floors except vertical in-wall drops (first floor) and rises (second floor) to fixtures. We assume the main trunk line is 1 inch until after the branch to bathroom 3, where it drops to ¾ inch. We also assume the branches to bathrooms 1, 2, 3 and the master bathroom are ¾ inch. We assume the branch to the laundry room is ½ inch, and that pipes in bathrooms 1, 2, and 3 drop to ½ inch after the first fixture encountered there. We assume the branch to the kitchen is ¾ inch, dropping to ½ inch after the first sink. We assume the vertical risers to all the master bathroom fixtures are ½ inch, from the main ¾ inch branch line.

For the case where the first floor is served by under-slab plumbing, all the second story fixture piping is assumed the same as for the pure above-ground trunk-and-branch system. Two separate under-slab branches are assumed, one to the kitchen, and one to bath 1. Both are assumed ¾ inch with series-connected fixtures typical of under-slab piping.

For the manifold system, actual measurements as shown in table 3-4 are used for pipe sizes and lengths. Volumes shown include the manifold and supply pipe from the water heater to the manifold.

Table A-2 compares the "water heater-to-fixture volumes" of the three different piping systems in the two-story house example.

Table A-2
Two Story House Example Pipe Volume Comparisons – Above Ground Trunk & Branch vs Under-Slab, vs Manifold System

FIXTURE	Above Ground T&B Volume WH-Fixt. (gal.)	Under-Slab 1st Floor + Above Ground T&B 2nd Floor WH-Fixt. (gal.)	Manifold WH-Fixt. (gal.)	RATIO Under-Slab/TB	RATIO Manifold/TB
Garage Sink	0.24	0.24	0.22	1	0.89
Bath 2 Sink	0.88	0.88	0.40	1	0.46
Bath 2 Sink	1.01	1.01	0.41	1	0.40
Bath 2 Tub/shower	1.10	1.10	0.68	1	0.62
Washing Machine	0.19	0.19	0.27	1	1.40
Laundry sink	0.22	0.22	0.34	1	1.53
Bath 3 Sink	0.92	0.92	0.38	1	0.41
Bath 3 Tub/shower	1.01	1.01	0.52	1	0.52
M. Bath Shower	0.96	0.96	0.43	1	0.44
M. Bath Sink	1.17	1.17	0.48	1	0.41
M. Bath Tub	1.31	1.31	0.85	1	0.65
M. Bath Sink	1.49	1.49	0.51	1	0.34
Bath 1 Sink	0.73	0.98	0.77	1.33	1.04
Bath 1 Tub/shower	0.96	1.14	0.50	1.19	0.52

Kitchen Sink 1	1.72	1.86	0.65	1.08	0.38
Kitchen Sink 2 + Dish W.	2.01	1.65	1.06	0.82	0.53

We see from table A-2 that the under-slab first floor piping benefits only the island sink in the kitchen. We also see that the manifold system generally has much less volume between the water heater and the fixtures than does the above-ground trunk-and-branch system, because of the smaller diameter pipes used. Exceptions are fixtures close to the water heater, where the trunk-and branch lines were also small-diameter lines and hence the extra volume of the supply line from the water heater to the distribution manifold negates the benefits of the manifold system serving those fixtures. The average ratio of "WH-to-fixture volume" for the manifold system divided by "WH-to-fixture volume" for the above-ground trunk-and-branch system was about 66%. Note, however, that the small diameter lines of the manifold system cool off to below a usable temperature very quickly – on the order of 2-3 minutes, and hence will result in purging the cooled-off water in the pipe to drain more frequently than for the trunk-and-branch system, especially if the latter is insulated. A more sophisticated analysis must be performed assuming realistic draw patterns and draw spacings before the various HWD systems can be fully compared.

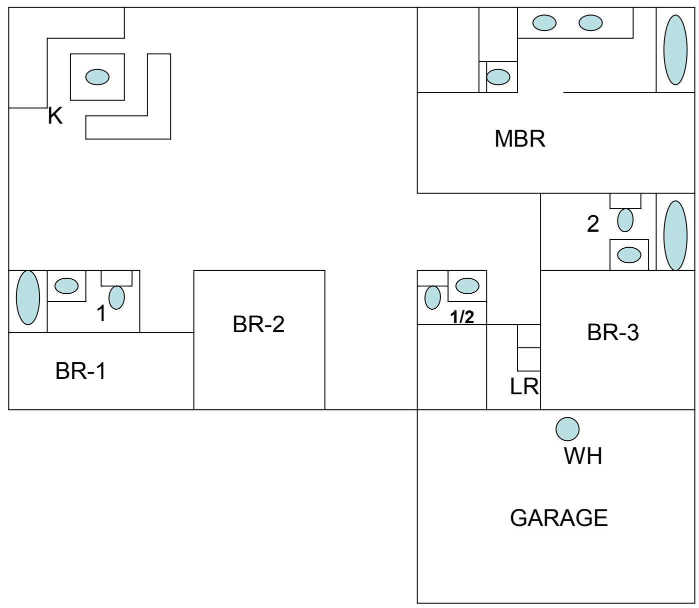


Figure A-1
Single Story House Example Plumbing Layout

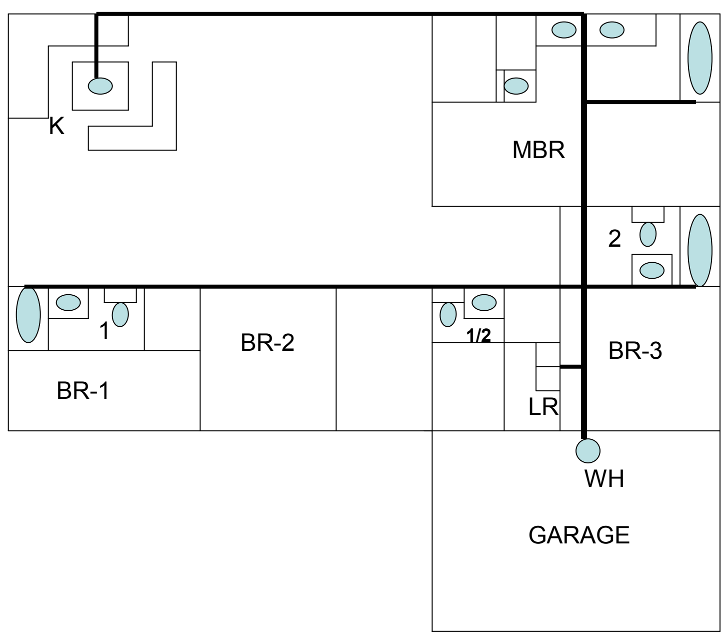


Figure A-2
Single Story House Example Trunk & Branch Piping Layout

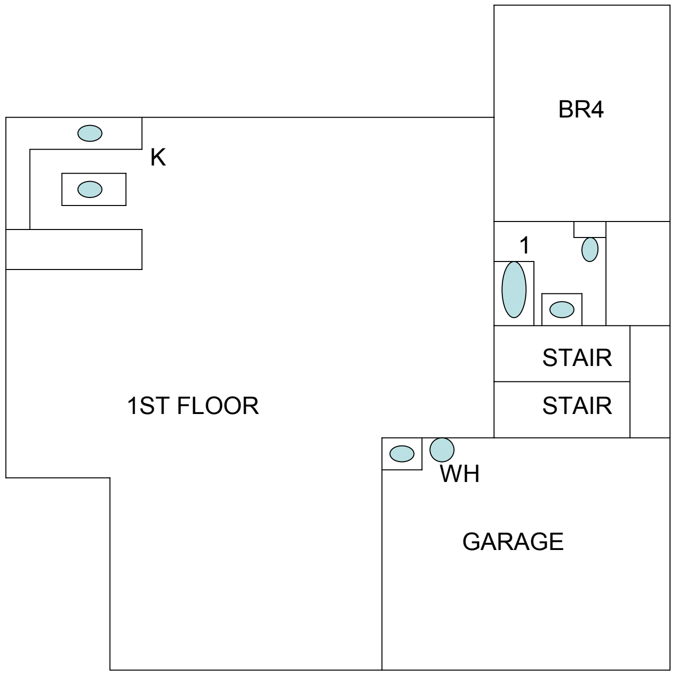


Figure A-3
Two-Story House Example Plumbing Layout, 1st Floor

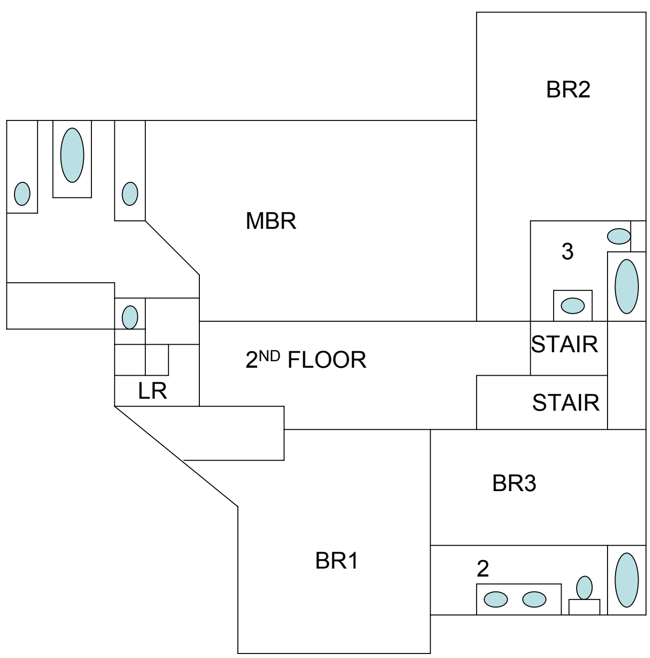


Figure A-4
Two-Story House Example Plumbing Layout, 2nd Floor

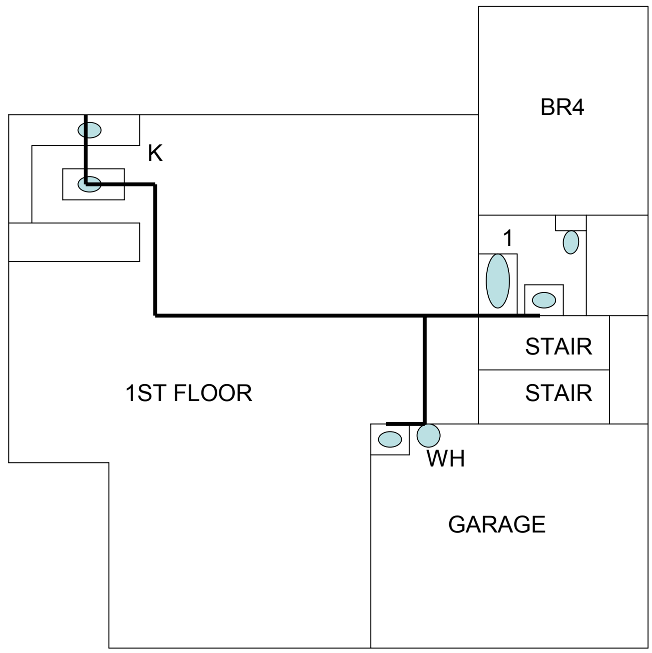


Figure A-5
Two-Story House Example Trunk & Branch Piping Layout, 1st Floor

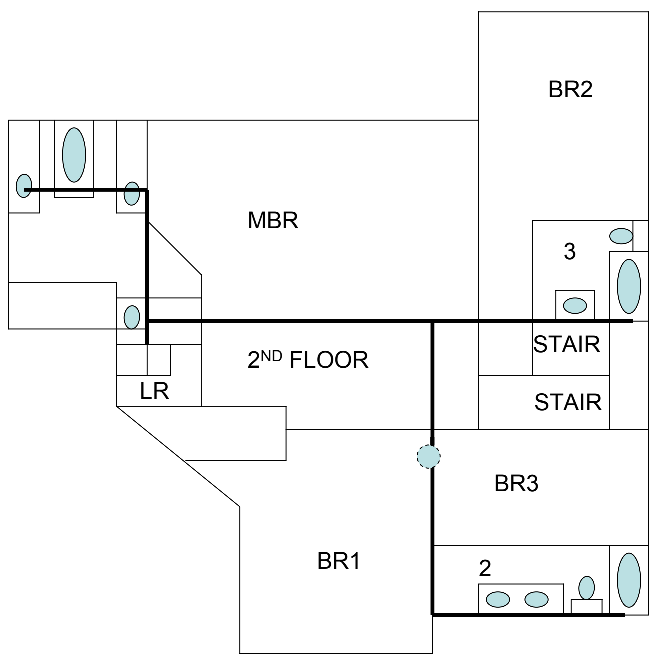


Figure A-6
Two-Story House Example Trunk & Branch Piping Layout, 2nd Floor

B
APPENDIX – LABORATORY TEST SETUP AND PROCEDURES

Test Laboratory

The Applied Energy Technology Water Heating Test Laboratory in Davis, California was moved to a larger facility specifically to accommodate the needs of this project. The new laboratory was large enough to allow testing of full-scale piping systems. Moreover, the 30 foot laboratory height was great enough to allow reasonable measurement of vertical piping performance characteristics as well as horizontal.

Hot water for draw tests was generated in the laboratory. Water was usually used once and dumped to drain. Initially hot water for the tests was provided by a single 80 gallon electric resistance water heater having a heating rate of 4500 Watts. As tests proceeded more water heaters were added to speed testing. As shown in figure B-1, ultimately three water heaters were connected in a series-parallel arrangement with valves and bypass lines that allowed them to be used individually or connected in any order. Total heating rate was ultimately approximately 10 kW, and was limited by the electrical feeder to the laboratory. Two 80 gallon tanks and one 50 gallon were used. The 50 gallon tank was run on 110 Volts instead of 220 to limit power draw on laboratory electrical circuits.

Data Collection Equipment and Procedures

A single data logger was used, having 25thermocouple and 16 thermistor channels. Three pulse-counting input channels were also available. One pulse-counting channel was used for the water flowmeter, another for the kWh meter. The water flow meter had a resolution of 0.0173 gallons per pulse. The kWh meter had a resolution of 1.2 Wh/pulse. Special fast-response immersion thermocouples were used in the tests of this project. Time response for the thermocouples was measured to range between 2.5 and 3.5 seconds, with most in the 3.0 to 3.3 second range, and accuracy was better than 0.1 F All sensors were calibrated prior to the beginning of testing.

Data was recorded and stored at 5 second intervals for all tests. This interval was a compromise between time-resolution of the data and the amount of data to be collected, stored, and manipulated.

During testing data was collected and stored internally in the data logger, which could store approximately 3 day's worth of continuous data in internal memory, and was also simultaneously downloaded and stored in a laboratory notebook computer which was connected to the data logger. Every day data was transferred from the laboratory notebook computer to an off-site central data processing computer. Multiple backup copies of both raw and processed data were made periodically and stored in separate locations in order to protect against catastrophic data loss in case of fire or other disaster.

Data Processing Procedures

Data generated by the data logger was downloaded in binary form to limit file size. After transferring to the central data processing computer the data files were converted to Excel spreadsheets. The first step in data processing was to "clean-up" data files by eliminating duplicate records and filling in missing data records (if possible) that sometimes occurred. Missing data scans could be regenerated from surrounding data by interpolation if they occurred during quasi-steady-state conditions. Missing data scans sometimes occurred during the peak transient portions of tests, which then invalidated that portion of the test.

Once a "clean" data file spreadsheet was created, a series of calculations were performed in the spreadsheet to produce the variety of performance results that were sought. Results from multiple tests were then manually transferred into a variety of data summary spreadsheets for comparison and results plotting.

Piping System Layout and Sensor Placement

Both horizontal and vertical piping tests were performed. For horizontal tests, piping was laid out in a serpentine pattern, with inlet and outlet on the same end, as shown in figures B-2, B-3, and B-4. The horizontal piping systems therefore had an even number of parallel straight sections. There were four passes for the ¾ inch rigid copper and PEX tests and six for the ½ inch rigid copper tests. As discussed in the accuracy resolution subsection of this section, longer piping lengths were required for smaller pipe diameters in order to maintain acceptable measurement accuracy. Piping total test section lengths varied from around 86 feet to slightly over 120 feet. See appendices C, D and E for exact dimensions. Temperature measurements were taken using immersion thermocouples placed at the U-bends in the serpentine piping, as well as at the inlet and outlet ends of the pipes. Flow adjustment valves were placed at the outlet ends of the test sections, accompanied by full-port ball valves that enabled rapid flow initiation in a step-change fashion. As seen in figure B-3, special valves and fittings on the inlet section of the pipe, ahead of the test section, allowed "priming" of inlet piping with fully hot water, so that draw tests could begin with a sharp temperature gradient, essentially a "square-wave" temperature flow into the test section at the time of valve opening.

The water flow meter was placed in the cold water line just upstream of the water heater tanks.

Air temperatures were measured on a plane approximately equal to that of the pipe, in three axial locations; the end nearest the tank, center, and far end of the serpentine piping. The air thermocouples were centered between pipes to minimize influence by radiation and convective heat transfer from the pipes.

For the vertical pipe tests, which were done on ¾ inch rigid copper piping, an approximately 21 foot long vertical pipe was placed perpendicular to the end of the horizontal test section. Figures B-5 and B-6 show the vertical pipe test configuration. The horizontal test section was insulated and was primed with hot water prior to the start of draws from the vertical test section. An isolation valve was installed at the base of the vertical test section to prevent natural convection flow between the horizontal and vertical pipe sections prior to test initiation. Separate tests were performed in order to observe the effects of this natural convection circulation. The reader should refer to the section of this report describing vertical pipe testing for more information.

Immersion Thermocouple Positioning

Initial tests were performed on ¾ inch rigid copper pipe, followed by ½ inch rigid copper pipe. As seen in figures B-2 and B-3, copper pipe T's were used, into which the immersion thermocouples were inserted. Side fitting lengths of the T's were adjusted such that the tips of the immersion thermocouples were inserted typically between 1/8 and ½ the diameter of the pipe. This was relatively easy to do in the ¾ inch piping, but more difficult in the ½ inch. As the piping diameter became smaller, the size of the thermocouple measuring tip (approximately 1/8 inch long) became large relative to the pipe diameter, as did the diameter of the thermocouple sheath (also 1/8 inch). It was difficult to insert the approximately 3-to-4 inch long thermocouple shafts straight enough in the ½ inch pipe side branches to avoid having the thermocouple sheaths (and sometimes the sensing tips) touch the side of the pipe T. Much trial and error was required to get the thermocouples inserted properly in the ½ rigid copper pipe.

The difficulty experienced with inserting immersion thermocouples in the ½ inch diameter pipe made it clear that a different thermocouple insertion system had to be used in smaller diameter pipes. Subsequently, a new technique was developed using a custom-fabricated low-mass compression seal clamping system that allowed direct insertion of the tips of the immersion thermocouples into the side of the pipe, without using a pipe T. Figure B-4 shows this technique as it was used in the PAX piping.

Test Procedures

A number of unexpected behaviors were observed during initial testing of each piping configuration. This lead to several modifications to the test apparatus and test procedures as testing proceeded. The final test apparatus configuration and procedures were as follows:

- A PVC plastic pipe coupling was installed on the inlet end of each piping test section, between the test section piping and the inlet ball valve. In some cases a PVC pipe T was used instead of a coupling in order to insert an immersion thermocouple just downstream of the inlet ball valve. This configuration can be seen in figure B-3. The purpose of using this plastic fitting was to prevent conduction heat transfer from the body of the ball valve into the test section piping while the inlet to the system was being primed with hot water. As seen in figure B-3, an insulated rubber hose brought hot water from the water heater to the inlet end of the test section. An insulated pipe T containing an immersion thermocouple was installed between the inlet rubber hose and the inlet ball valve. Between the thermocouple T and the inlet ball valve was another pipe T and side-branch bypass ball valve that allowed purging water from the inlet section in order to prime it with hot water. This inlet side bypass valve also made it possible to use pumps and /or cold water flushing to precondition the temperature of the test section.

- The inlet rubber hose had to enter vertically above the test section, to prevent natural convection flow of cold water in the test section back into the inlet hose during the few seconds between the opening of the inlet ball valve and the opening of the discharge ball valve.

- Typical water draw flow rates tested were 0.5, 1.0, 2.0, 3.0, 4.0 and 5.0 gallons per minute (gpm). Initially these flow rates were adjusted between each test. Ultimately, an outlet valve "tree" was installed, having four flow adjust valves preset to four different flow rates, each accompanied by a fast-acting ball-valve for flow initiation. This valve tree is seen on the outlet end of the test section in figure B-3. The four preset flow rates were 0.5, 1.0, 2.0, and 3.0 gpm. By opening multiple valves simultaneously in various combinations, this made it possible to achieve flow rates in approximately ½ gpm increments from 0.5 gpm through 5.0 gpm if desired.

- Hot water temperature, initial test section pipe temperature, and room air temperature could all be controlled independently. Room air temperature was controlled indirectly by performing tests at different times of day. Fortunately, the climate in Davis, California, where the test laboratory is located, regularly experiences 30 to 40 F temperature changes between day and night, enabling similar changes in laboratory air temperature. Entering hot water temperature was controlled by the tank thermostat settings. Initial test section piping temperature was controlled by one of several means, depending on the temperatures desired in the test section and the time of year. During winter months, entering cold water could be used to purge the test section and reestablish a cold initial pipe temperature before each test. During late summer months, ice and a recirculating pump were used to precondition the test section temperature to temperatures lower than entering cold water temperature. To achieve medium (e.g. 90 F) initial test section temperatures, one of two methods was used. One method was to initiate a maximum flow-rate hot water draw in the test section to raise the entire test section to a fairly high but uniform temperature, and then let the test section cool off on its own down to a desired initial test section temperature. Another method was to use a recirculation pump combined with hot and/or cold water draws into the piping system in order to establish a desired initial test section temperature.

- A typical test consisted of first establishing the desired entering hot water temperature by either adjusting the tank thermostat(s), or manually controlling power to the tanks, shutting the power off to the downstream (exit) tank when the desired temperature was reached. The room air temperature was adjusted by observing lab air temperature, and performing tests at various times of day to achieve different air temperatures. Next, the desired initial test section piping temperature was established. Finally, the test section inlet and outlet ball valves were closed, and the entry section was primed by dumping hot water to drain through the inlet bypass valve until temperature was constant within about 0.2 F. This usually took about 2-3 minutes. Upon reaching a constant inlet temperature, the purge valve was closed and the test section inlet ball valve was immediately opened, followed as closely as possible (typically 3 seconds or less) by opening of the desired outlet ball valve(s).

- Most tests consisted of three distinct phases, as shown in figure B-7. The first phase was the "delivery" phase while hot water was traversing the test section piping until usably hot water was obtained at the outlet. Data from this phase of the tests was used to develop information on time, water, and energy waste while waiting for "hot-enough-to-use" water to arrive at fixtures. The second phase, which proceeded continuously from the first, was the "use" phase, where water was drawn until test section outlet temperature had remained relatively constant for at least 3 minutes, after which flow was terminated. The water steady-state temperature drop from inlet to outlet that was measured during this phase was used to calculate piping heat loss (UA) factors as a function of flow rate and other parameters. The third phase was piping cool-down, where test section piping temperature was observed as the piping cooled. The cool-down phase was usually allowed to proceed at least 10 minutes, and sometimes up to 24 hours. Combined with information on total pipe+water+insulation thermal mass, data from this phase was used to determine piping heat loss (UA) factors under zero flow conditions.

Results Calculation Procedures

Laboratory testing produced three important data outputs as a function of relevant parameters. Other important results can be calculated from these primary data outputs. The three primary data outputs were AF/PV ratio, flowing UA value, and zero-flow UA value.

The delivery-phase of each test was used to measure the amount of water wasted to drain before "usably hot" water (defined for test purposes as 105 F) was obtained, under a variety of different test conditions. It was learned that the best way to present the highest resolution information on water waste (and hence energy waste) was to present the ratio of actual wasted or actual flow gallons (AF) divided by pipe volume (PV), or AF/PV ratio, as a function of relevant parameters. While time spent waiting for this water to arrive was also directly measured, it is more convenient to use a known flow rate and measured AF/PV ratios to calculate what the time spent waiting for hot water was under different conditions. Hence separate correlations for wait time were not developed.

The steady-state use-phase of the test water draws was used to determine piping heat loss or $UA_{flowing}$ factors under flowing water conditions. Draws were continued until test section inlet and outlet temperatures were relatively constant for a period of typically at least 3 minutes. The temperature drop thus measured from one temperature measuring station to another could then be used to calculate the piping heat loss $UA_{flowing}$ factor. Due to the mass of the valve tree on the outlet end of the test section, the temperature drop through the test section was calculated based on either the temperature of the next temperature measuring station back from the end, or in later tests from a separate thermocouple inserted near the outlet end of the test section, but away from the thermally massive valve tree. $UA_{flowing}$ was calculated from the equation:

$$Q = UA_{flowing}(T_{pipe\ avg.} - T_{air}) = (\text{mass flow rate})(Cp)(T_{hot\ in} - T_{hot\ out})$$

Where Q = heat loss rate

Cp = specific heat of water

$T_{pipe\ avg.}$ was calculated using log-mean temperature differences, rather than using a simple arithmetic average (a more correct method of calculation than simple average – see heat transfer textbooks for more information)

Since Cp for water is known, and flow rate and temperatures were measured, the only unknown was $UA_{flowing}$, which could be calculated.

Ideally, $UA_{flowing}$ is a not a function of flow rate or temperatures, and is a constant for practical purposes. However, test results showed that $UA_{flowing}$ was in fact somewhat a function of temperatures and flow rate, but within a limited range – see the detailed test results. A constant value can be assumed for most engineering calculations.

The cool-down phase of the tests was used to calculate the pipe heat loss $UA_{zero\text{-}flow}$ factor under zero-flow conditions. The procedure for doing this is different than under flowing conditions. Under zero-flow conditions, the rate of temperature drop of the water in the pipe vs time is used to determine the $UA_{zero\text{-}flow}$ factor using the equation:

$$Q_{cd} = (mCP)_{wpi}(T_{pipe\ time\ 1} - T_{pipe\ time\ 2})/(\Delta t) = UA_{zero\text{-}flow}(T_{pipe\ avg.} - T_{air\ avg.})$$

Where
Δt = time period over which temperature drop was measured
Q_{cd} = pipe heat loss rate during cool-down
$(mCp)_{wpi}$ = the total mass times specific heat of the water in the pipe plus the pipe material plus the insulation material = (m water)(Cp water)+(m pipe)(Cp pipe)+(m insul.)(Cp insul.) per foot of pipe

$T_{pipe\ average}$ was calculated using log-mean temperature differences vs time, rather than using a simple arithmetic average (a more correct method of calculation than simple average).

For copper pipe the specific heat was readily available and the mass of pipe could be readily calculated from knowledge of pipe dimensions and density of copper.

For the foam insulation, density was calculated based on weight and volume measurements on a section of insulation. Specific heat of the insulation was assumed to be the same as for the parent polyethylene material.

For PAX piping, which is a 3-layer composite of high density cross-linked polyethylene (PEX), aluminum, and another layer of PEX, mass and effective specific heat were estimated from densities and specific heats of the parent materials and measurements of layer thicknesses from a section of the pipe used in the test laboratory.

Once the mass and specific heats were known, the measurements of temperatures vs time were used to solve for the only unknown – $UA_{zero\text{-}flow}$. Again ideally $UA_{zero\text{-}flow}$ is a constant, independent of temperatures and time. However, the test data showed that $UA_{zero\text{-}flow}$ did in fact vary somewhat with temperatures. Reasonable results can be obtained, however, by assuming a constant value, as shown in the detailed results sections. Note that $UA_{zero\text{-}flow}$ was calculated at short (one minute) intervals, and then the average over many time intervals was taken. In this way we could see variation of cool-down $UA_{zero\text{-}flow}$ with time and we did not need to assume an exponential decay function to calculate $UA_{zero\text{-}flow}$. We could have just as easily used an exponential decay function. In fact, when calculating the temperature to which a pipe has cooled as a function of time, it is easiest to use the above-measured $UA_{zero\text{-}flow}$ values in an exponential decay function of the form:

$(T_{pipe\ time\ 2} - T_{air})/(T_{pipe\ time\ 1} - T_{air}) = e^{-(UA/mCp)t}$.

Where:

mCp = total mass times specific heat of the pipe+water+insulation
t = time in proper units to make the expression (UA/mCp)(t) dimensionless

The temperature of water in the pipe, and the energy loss due to cool-down, can be calculated at any time after flow has stopped by using the measured $UA_{zero\text{-}flow}$ value, the piping system mass and specific heat per foot, and assuming an initial pipe temperature and surrounding air temperature.

Horizontal Piping System Test Matrix

For the horizontal in-air piping tests, tests were performed at usually 5 flow rates, two to four air temperatures, two hot water temperatures, and two initial pipe temperatures. This means that a minimum of (5)(2)(2)(2) = 40 tests were performed for each piping configuration, and usually closer to 50. Additionally, due to test variability, it was learned that each test had to be repeated at least 3 times in order to ensure that the test result was not anomalous due to some unforeseen test or measurement error, or unusual transient behavior. This meant that a total of approximately 150 tests had to be run for each pipe configuration. For example 150 tests would be performed on bare ½ inch copper pipe, another 150 tests on ½ inch copper pipe with ½ inch foam insulation, and another 150 tests on ½ inch copper pipe with ¾ inch foam insulation. Due to slight variations in flow rates and temperatures in each test, what resulted was a fairly comprehensive set of performance results that could be correlated with respect to important influencing parameters. The flow rates tested were typically around 0.5, 1.0, 2.0, 3.0, and 4.0 gpm. The hot water temperatures tested were typically around 135 F and 115 F. The initial pipe temperatures tested were typically around 65 F and 90 F. The room air temperatures tested were typically around 50, 60, 70, 80, and 90 F, depending on time of year and time of day. Temperatures of 60-75 F could be achieved year around.

Accuracy and Error Bounds Limitations

An analysis of accuracy limitations with the test setup has shown two primary limiting factors:

- The minimum water flow meter reading of 0.0173 gallons per pulse, while excellent, limits the accuracy with which flows can be measured in shorter pipe lengths, especially with smaller diameter pipes. One pulse of the flow meter represents over 3% of the volume of the first 20 foot section of ¾ inch rigid copper pipe. That increases to around 8% of the first 20 foot section for ½ inch rigid copper pipe, and would be approximately 15% of the volume of the first 20 feet of a 3/8 inch diameter pipe. Said another way, it becomes increasingly difficult to accurately detect the propagating temperature front in the beginning sections of the pipe during draws as the pipes get smaller. Using longer pipe lengths lets us see the shape of the temperature front more clearly.

- The standard data storage interval used in the tests is 5 seconds. While this is adequate for most purposes, at high flow rates and smaller pipe diameters, temperatures rise very steeply in 5 seconds, especially in shorter pipe lengths. This limits the ability to accurately determine the time and gallons of flow at which "acceptably warm" (defined as 105 F for test purposes) water reaches early stations along the pipe. The solution is to recognize that results at short pipe lengths may not be accurate, and to examine mostly data from longer pipe lengths. Fortunately, it is probably true that at shorter pipe lengths in smaller diameter pipe, flow proceeds in a nearly "square-wave" plug-flow manner anyway, developing a temperature distribution mostly at longer pipe lengths.

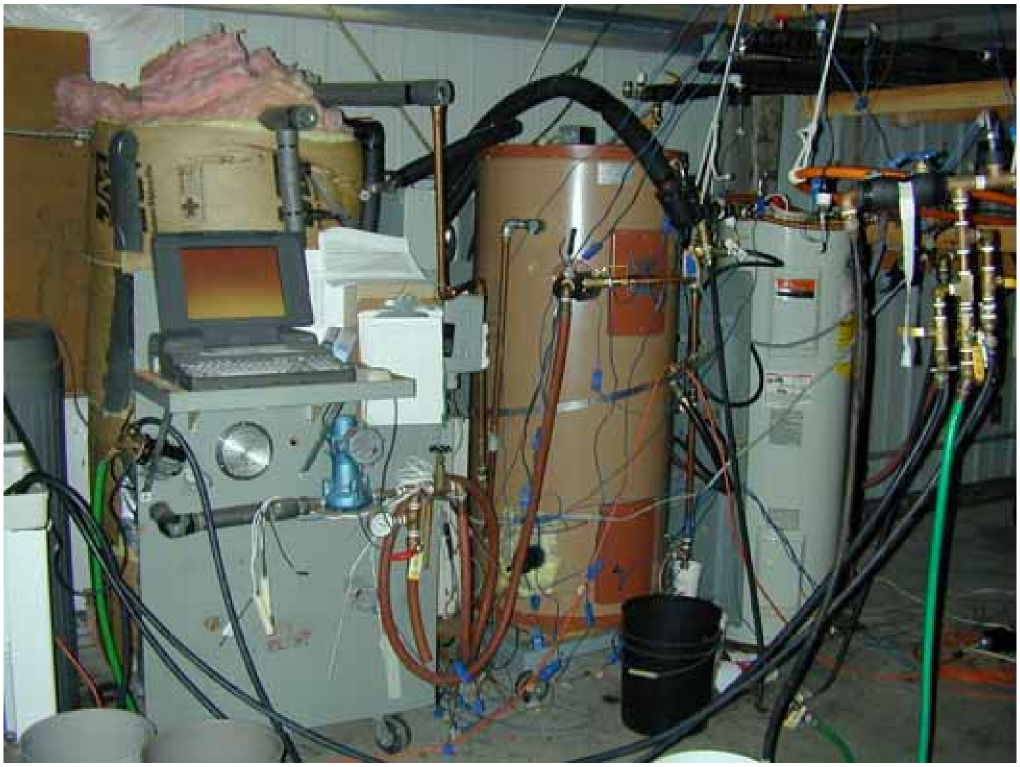


FIGURE B-1
THREE TANKS IN LABORATORY TEST SYSTEM

FIGURE B-2
SERPENTINE PIPING LAYOUT – ½ INCH COPPER ABOVE, ¾ INCH COPPER (INSULATED) BELOW

FIGURE B-3
SERPENTINE PIPING LAYOUT – NOTE ISOLATION COUPLER AND FLOW ADJUST VALVE TREE

FIGURE B-4
¾ PAX PIPING LAYOUT – NOTE CUSTOM THERMOCOUPLE MOUNTING

FIGURE B-5
¾ INCH COPPER VERTICAL PIPING LAYOUT – LOWER

FIGURE B-6
¾ INCH COPPER VERTICAL PIPING LAYOUT – UPPER

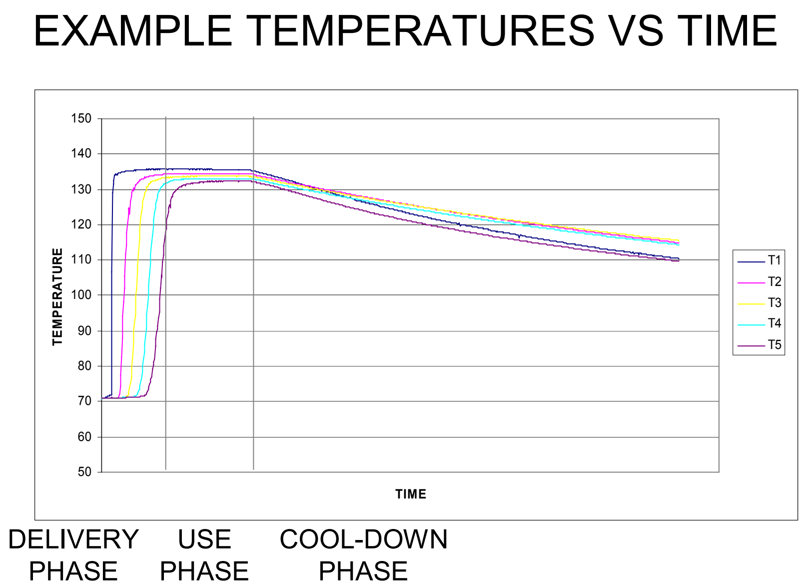


FIGURE B-7
TYPICAL TEST RESULT SHOWING DELIVERY, USE, AND COOL-DOWN PORTIONS

C
APPENDIX – ¾ INCH RIGID COPPER PIPE TESTS - DETAILED DATA

Technical information on the ¾ inch rigid copper piping system was as follows:

Table C-1
¾ Inch Rigid Copper Piping Technical Data

	WATER	PIPE	½ FOAM (R-2.9)	¾ FOAM (R-4.7)
ID (inches)	-	0.785	0.875	0.875
OD (inches)	0.785	0.875	1.875	2.375
Density(lbm/ft^3)	62.0*	559	0.58	0.58
Cp (Btu/lbm F)	1.0	0.0915	0.48	0.48
mCp (Btu/ft F)	0.2084	0.0417	0.0042	0.0074

*Water density at 100 F

Figure C-1
¾ Inch Bare Rigid Copper Test Section

Figure C-2
¾ Inch Insulated Rigid Copper Test Section

Table C-2
UA Summary Table – ¾ Inch Bare Rigid Copper Pipe

FLOW RATE (GPM)	**UA (BTU/HR FT F)**	**T PIPE (F)**	**T AIR (F)**	**TEMPERATURE DIFFERENCE (F)**
0	0.35059	110.6	78.8	31.8
0	0.38049	112.4	81.2	31.2
0	0.31933	133.8	86.7	47.1
0	0.33145	133.6	84.0	49.6
0	0.3979	124.9	76.8	48.1
0	0.4595	109.3	61.8	47.5
0.5	0.4117	114.2	67.5	46.7
0	0.3164	132.4	77.0	55.4
0	0.3349	129.3	71.4	57.9
0	0.3984	125.0	68.6	56.4
0	0.4172	126.5	66.6	59.9
0.46	0.4152	113.6	62.4	51.2
0	0.3974	128.5	63.7	64.8
0	0.4037	127.0	64.6	62.4
0	0.4320	128.0	64.0	64.0
0.44	0.4189	131.5	69.3	62.2
0.75	0.3960	133.4	69.2	64.2
1.06	0.3981	133.4	69.6	63.8
0	0.4122	124.4	56.2	68.2
0	0.4209	129.9	59.0	70.9
0	0.4359	132.0	63.4	68.6
1.69	0.4325	134.5	67.5	67.0
1.8	0.4384	133.7	64.5	69.2
2.39	0.4515	134.2	63.7	70.5
2.68	0.4384	135.1	64.3	70.8
3.17	0.4144	135.0	65.8	69.2
4.19	0.4112	135.1	64.1	71.0

Table C-3
UA Summary Table – ¾ Inch Rigid Copper Pipe With R-2.9 Insulation

FLOW RATE (GPM)	UA (BTU/HR FT F)	T PIPE (F)	T AIR (F)	TEMPERATURE DIFFERENCE (F)
0	0.1435	102.0	70.4	31.6
0	0.1498	132.2	79.2	53.0
3.65	0.2169	133.2	78.4	54.8
0	0.1526	129.4	70.3	59.1
0	0.1542	132.8	75.1	57.7
0.63	0.1902	130.0	70.3	59.7
0	0.1466	131.8	67.5	64.3
0	0.1489	129.7	67.1	62.6
0	0.1513	132.2	67.4	64.8
0	0.1528	131.6	70.5	61.1
0	0.1535	131.5	66.7	64.8
0	0.1570	129.4	68.9	60.5
0	0.1585	132.3	71.6	60.7
1.77	0.2078	130.6	67.7	62.9
2.87	0.2456	133.6	70.1	63.5
0	0.1481	131.5	65.8	65.7
0	0.1486	133.3	65.2	68.1
0	0.1500	132.4	67.1	65.3
0.62	0.1886	133.5	65.0	68.5
0.85	0.1937	135.0	65.4	69.6
1.0	0.2020	132.8	66.7	66.1
1.01	0.2078	133.2	66.9	66.3
1.47	0.2324	132.9	65.1	67.8
1.54	0.2387	133.7	66.5	67.2
3.36	0.2572	134.0	67.3	66.7

Table C-4
UA Summary Table – ¾ Inch Rigid Copper Pipe With R-4.7 Insulation

FLOW RATE (GPM)	UA (BTU/HR FT F)	T PIPE (F)	T AIR (F)	TEMPERATURE DIFFERENCE (F)
0	0.1357	113.6	83.5	30.1
0	0.1436	115.2	79.7	35.5
2.14	0.1341	115.6	77.8	37.8
0	0.1368	114.5	73.1	41.4
0	0.1478	133.6	89.3	44.3
1.0	0.1794	115.3	71.1	44.2
0	0.1340	113.9	65.5	48.3
0	0.1345	114.7	66.7	48.1
0	0.1454	132.1	84.2	47.9
0	0.1467	134.3	84.9	49.4
0	0.1476	130.9	84.9	46.0
0	0.1502	131.7	82.7	49.0
0	0.1572	133.7	83.7	49.9
0.42	0.1686	114.9	65.1	49.7
0.45	0.1760	135.6	85.7	49.9
1.76	0.2252	133.1	85.4	47.7
4.34	0.1895	132.2	82.6	49.6
1.07	0.1751	135.6	84.4	51.2
2.16	0.1747	132.3	81.8	50.5
0	0.1374	125.0	66.8	58.2
0	0.1529	128.7	72.4	56.3
0	0.1376	133.4	68.9	64.5
0.75	0.1427	132.3	71.0	61.3
1.05	0.1800	131.3	69.3	62.0
0	0.1362	132.0	65.5	66.5
0	0.1364	134.5	67.2	67.3
0	0.1368	132.0	64.1	67.9
0	0.1384	132.7	67.2	65.5
0	0.1385	131.2	66.1	65.1
0	0.1397	134.6	67.7	66.9
0	0.1443	133.0	63.6	69.4
0	0.1446	133.5	65.7	67.8
0.44	0.1561	134.4	66.4	68.0

0.47	0.1440	134.4	69.3	65.1
0.62	0.1652	134.6	66.2	68.4
0.95	0.1882	134.6	67.8	66.8
1.03	0.1886	134.1	67.0	67.1
1.09	0.1739	135.6	67.4	68.2
1.09	0.1859	135.1	65.6	69.5
1.09	0.1977	133.9	67.3	66.6
1.27	0.2226	136.1	68.1	68.0
2.51	0.2392	135.9	68.1	67.9
3.15	0.2584	133.3	64.5	68.8
0	0.1405	134.4	63.1	71.3
0.51	0.1492	135.3	63.6	71.7
1.0	0.1865	135.9	63.4	72.5
1.03	0.1944	135.9	62.9	73.0

Table C-5
UA Summary Table – ¾ Inch Rigid Copper Pipe Curve Fits to High Data

FLOW RATE (GPM)	UA BARE (BTU/HR FT F)	UA R-2.9 (BTU/HR FT F)	UA R-4.7 (BTU/HR FT F)
0	0.388	0.15	0.142
0.5	0.415	0.178	0.17
1.0	0.44	0.208	0.22
1.5	0.44	0.235	0.24
2.0	0.44	0.25	0.24
3.0	0.44	0.25	0.24
4.0	0.44	0.25	0.24
5.0	0.44	0.25	0.24

Table C-6
AF/PV Summary Table – ¾ Inch Bare Rigid Copper Pipe

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.7 FT. (0.5454 gal.)	AF/PV AT L=43.5 FT. (1.093 gal.)	AF/PV AT L=65.2 FT. (1.6395 gal.)	AF/PV AT L=86.3 FT. (2.169 gal.)
0.42	110.8	78.2	84.6	0.178	1.879	1.553	1.504	1.561
0.45	114.1	62.9	62.5	0.178	1.759	1.531	1.508	1.564
0.89	114.4	65.5	66.8	0.192	1.603	1.428	1.365	1.372
1.03	113.7	65.7	67.1	0.181	1.560	1.408	1.373	1.383
1.04	114.2	66.8	81.4	0.194	1.544	1.418	1.361	1.369
2.05	113.6	66.4	74.5	0.182	1.564	1.402	1.357	1.352
2.50	113.5	63.7	79.0	0.171	1.613	1.434	1.350	1.366
3.53	114.2	65.7	76.9	0.190	1.535	1.349	1.338	1.335
4.20	113.2	63.2	79.1	0.164	1.607	1.431	1.358	1.362
4.61	114.2	66.1	75.2	0.191	1.531	1.388	1.310	1.298
5.85	112.5	63.5	78.9	0.153	1.603	1.365	1.373	1.317
0.46	114.6	67.4	81.9	0.203	1.658	1.491	1.437	1.445
1.72	117.8	64.4	67.3	0.240	1.465	1.358	1.314	1.324
2.09	117.3	64.7	71.7	0.234	1.483	1.367	1.314	1.324
4.60	117.0	65.3	71.8	0.232	1.340	1.327	1.303	1.305
4.70	115.3	65.3	75.2	0.206	1.454	1.324	1.321	1.326
0.58	114.3	79.2	83.4	0.265	1.495	1.404	1.357	1.378
1.98	112.3	86.0	88.5	0.277	1.423	1.339	1.300	1.305
0.41	113.5	86.1	86.9	0.310	1.682	1.401	1.357	1.395
0.97	113.4	86.7	88.1	0.315	1.435	1.348	1.306	1.318
3.14	114.3	86.8	88.0	0.338	1.287	1.273	1.231	1.260
3.83	113.7	86.3	88.5	0.317	1.388	1.282	1.250	1.258
0.50	132.2	59.0	59.2	0.371	1.381	1.327	1.311	1.316
0.50	131.1	58.9	58.0	0.361	1.434	1.332	1.306	1.316
0.99	134.6	60.4	58.6	0.399	1.308	1.269	1.255	1.274
1.98	134.4	60.5	58.8	0.398	1.260	1.243	1.229	1.260
3.39	134.1	59.9	58.2	0.392	1.292	1.228	1.231	1.243
4.46	133.9	61.0	58.2		1.239	1.240	1.212	1.211
0.47	134.9	61.4	58.6	0.407	1.361	1.323	1.317	1.321
0.54	134.4	67.6	80.8	0.440	1.369	1.285	1.269	1.275
0.56	134.1	64.1	63.1	0.415	1.354	1.283	1.276	1.294
1.02	132.2	64.7	83.8	0.403	1.358	1.281	1.258	1.274
1.02	134.1	63.7	59.3	0.413	1.324	1.279	1.253	1.263

1.04	135.1	66.4	62.5	0.438	1.320	1.266	1.244	1.265
1.24	127.2	74.2	76.2	0.418	1.295	1.251	1.240	1.262
1.86	133.0	64.7	64.5	0.410	1.262	1.264	1.242	1.258
2.04	131.5	65.9	85.2	0.404	1.278	1.251	1.229	1.248
2.08	134.9	63.7	62.2	0.420	1.267	1.238	1.224	1.241
2.42	133.5	66.2	63.8	0.423	1.244	1.238	1.226	1.246
2.67	134.5	68.3	64.3	0.446	1.211	1.224	1.221	1.231
2.94	134.2	62.7	61.7	0.408	1.193	1.231	1.224	1.241
3.16	132.2	64.4	85.4	0.401	1.270	1.257	1.225	1.247
3.18	134.4	67.6	66.6	0.440	1.165	1.233	1.206	1.227
4.13	134.7	67.3	64.0	0.441	1.197	1.213	1.191	1.207
1.70	133.9	69.9	68.8	0.452	1.215	1.246	1.236	1.248
3.36	133.2	71.2	75.2	0.455	1.163	1.216	1.205	1.213
4.24	134.5	75.2	81.1	0.497	1.157	1.180	1.161	1.185
3.22	134.5	76.9	85.0	0.512	1.134	1.187	1.181	1.211
0.55	134.7	84.7	62.7	0.593	1.206	1.213	1.214	1.229
1.03	134.9	84.7	62.5	0.596	1.185	1.198	1.195	1.210
1.97	131.0	86.6	84.5	0.585	1.187	1.168	1.165	1.192
2.03	134.4	84.8	62.5	0.593	1.189	1.174	1.187	1.194
3.02	133.1	84.7	61.9	0.581	1.153	1.172	1.170	1.192
3.10	132.0	86.0	85.9	0.586	1.139	1.142	1.167	1.184
0.50	135.3	86.3	79.7	0.618	1.197	1.176	1.174	1.189
0.97	133.1	86.6	82.3	0.604	1.228	1.194	1.192	1.211

Table C-7
AF/PV Summary Table – ¾ Inch Rigid Copper Pipe With R-2.9 Insulation

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.7 FT. (0.5454 gal.)	AF/PV AT L=43.5 FT. (1.093 gal.)	AF/PV AT L=65.2 FT. (1.6395 gal.)	AF/PV AT L=86.3 FT. (2.169 gal.)
0.60	129.7	66.2	69.8	0.389	1.355	1.286	1.262	1.269
1.77	130.9	65.6	67.3	0.396	1.296	1.270	1.267	1.273
0.84	134.5	66.5	64.6	0.434	1.319	1.260	1.250	1.254
0.96	132.3	67.0	65.7	0.418	1.331	1.274	1.252	1.274
0.98	132.6	69.9	66.8	0.440	1.289	1.248	1.237	1.245
3.10	131.9	68.8	69.0	0.426	1.250	1.217	1.230	1.238
4.08	134.0	68.4	67.0	0.442	1.237	1.235	1.219	1.227
0.54	133.0	70.8	64.2	0.450	1.274	1.257	1.239	1.258
2.89	133.7	71.5	70.1	0.462	1.189	1.207	1.218	1.236
3.34	134.1	69.4	67.4	0.450	1.265	1.217	1.226	1.233
3.61	133.4	74.3	78.6	0.481	1.241	1.210	1.199	1.210
4.17	133.6	71.2	71.8	0.458	1.185	1.211	1.188	1.205
4.36	134.1	72.6	72.8	0.473	1.082	1.145	1.192	1.213

Table C-8
AF/PV Summary Table – ¾ Inch Rigid Copper Pipe With R-4.7 Insulation

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.7 FT. (0.5454 gal.)	AF/PV AT L=43.5 FT. (1.093 gal.)	AF/PV AT L=65.2 FT. (1.6395 gal.)	AF/PV AT L=86.3 FT. (2.169 gal.)
0.41	114.3	62.1	65.8	0.178	1.799	1.533	1.465	1.464
0.99	114.9	64.5	70.9	0.196	1.593	1.423	1.364	1.356
2.16	115.6	62.5	77.5	0.199	1.508	1.389	1.339	1.338
2.18	131.9	63.0	81.4	0.390	1.303	1.241	1.241	1.259
5.57	130.7	66.4	70.7	0.399	1.215	1.213	1.173	1.230
0.43	134.1	63.9	66.0	0.415	1.281	1.262	1.252	1.250
0.92	133.9	67.7	67.5	0.436	1.299	1.249	1.232	1.251
0.97	135.1	64.1	63.0	0.424	1.329	1.269	1.262	1.267
1.06	134.9	63.5	65.7	0.419	1.286	1.277	1.260	1.267
1.08	133.2	70.2	67.2	0.448	1.270	1.241	1.223	1.244
3.19	132.8	67.6	65.5	0.426	1.278	1.238	1.225	1.232
0.45	135.0	71.1	85.0	0.469	1.304	1.265	1.250	1.239
1.30	135.2	68.8	67.7	0.455	1.239	1.261	1.244	1.260
1.79	132.8	72.5	85.2	0.461	1.205	1.223	1.215	1.238
1.05	135.1	78.0	84.3	0.527	1.209	1.230	1.211	1.223
1.06	134.1	77.4	83.3	0.513	1.245	1.225	1.219	1.237
4.30	132.0	85.1	82.7	0.575	1.116	1.144	1.138	1.157

D

APPENDIX – 1/2 INCH RIGID COPPER PIPE TESTS - DETAILED DATA

Technical information on the 1/2 inch rigid copper piping system was as follows:

Table D-1
1/2 Inch Rigid Copper Piping Technical Data

	WATER	PIPE	½ FOAM (R-3.1)	¾ FOAM (R-5.2)
ID (inches)	-	0.545	0.625	0.625
OD (inches)	0.545	0.625	1.625	2.125
Density(lbm/ft^3)	62.0*	559	0.58	0.58
Cp (Btu/lbm F)	1.0	0.0915	0.48	0.48
mCp (Btu/ft F)	0.1004	0.02611	0.0034	0.0063

- Water density at 100 F

Figure D-1
½ Inch Bare Rigid Copper Test Section

Figure D-2
½ Inch Insulated Rigid Copper Test Section (Upper pipe)

Table D-2
UA Summary Table – 1/2 Inch Bare Rigid Copper Pipe

FLOW RATE (GPM)	UA (Btu/hr ft F)	T pipe (F)	T air (F)	TEMP. DIFF. (F)
0	0.1977	114.5	82.5	32.0
0	0.2171	126.1	99.0	27.1
0	0.2173	113.8	83.0	30.8
0	0.2181	114.7	84.0	30.7
0.45	0.2696	115.7	82.2	33.5
0.45	0.3034	114.5	84.1	30.4
0.46	0.2771	116.6	84.5	32.1
0.47	0.2721	129.6	99.5	30.1
0.47	0.2758	115.7	81.0	34.7
0.96	0.2773	115.7	81.8	33.9
1.01	0.2849	116.5	84.5	31.9
1.79	0.2811	116.1	84.4	31.7
2.95	0.2906	115.7	84.0	31.7
0	0.1996	115.6	79.9	35.7
0	0.2084	114.6	76.6	38.1
0	0.2109	113.2	73.9	39.3
0.46	0.2753	117.0	78.9	38.1
0.46	0.2887	117.4	77.7	39.7
2.05	0.3089	115.7	75.9	39.8
0	0.2127	115.4	74.4	41.1
0	0.2286	115.1	71.1	44.0
0.44	0.2922	116.2	71.3	44.9
0.45	0.2872	115.8	72.4	43.4
0.98	0.2995	115.9	73.1	42.7
1.08	0.3104	115.5	74.1	41.4
1.82	0.3079	114.8	74.1	40.7
2.13	0.3202	117.0	73.7	43.3
4.25	0.3318	116.3	74.1	42.2
0	0.2399	116.9	70.0	46.9
0.46	0.3011	116.5	70.5	46.0
1.03	0.3115	116.1	70.6	45.4
1.70	0.3274	116.3	70.4	45.9
2.82	0.3306	116.1	70.6	45.6
0	0.2172	126.0	72.2	53.8
0	0.2211	126.4	71.8	54.7
0	0.2254	113.9	63.6	50.2

0	0.2400	113.9	60.4	53.5
0	0.2442	114.6	60.2	54.4
0.40	0.3159	115.3	60.4	54.9
0.45	0.3170	115.2	60.4	54.8
0.46	0.3030	121.5	69.7	51.8
0.99	0.3048	121.2	69.7	51.5
1.60	0.3097	120.4	70.0	50.5
0	0.2657	129.1	70.1	59.0
0.44	0.3235	131.1	72.9	58.2
0.97	0.3176	115.5	60.4	55.1
1.77	0.3332	115.6	60.3	55.4
3.05	0.3451	115.8	60.3	55.5
3.80	0.3310	115.8	60.3	55.5
0.40	0.3112	133.5	70.0	63.5
0.42	0.3000	133.2	71.6	61.6
1.48	0.3157	132.5	69.6	62.9
2.56	0.3052	131.3	69.7	61.6
0	0.2416	130.8	64.9	65.8
0	0.2464	132.7	65.6	67.1
0	0.2478	133.3	65.8	67.5
0.46	0.3250	135.2	66.0	69.1
0.47	0.3160	134.7	68.6	66.1
0.48	0.3192	133.8	68.6	65.2
0.93	0.3410	136.3	65.8	70.5
1.55	0.3464	136.0	66.1	69.9
1.55	0.3510	135.6	65.8	69.8
2.97	0.3678	136.5	65.5	71.0

Table D-3
UA Summary Table – 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation

FLOW RATE (GPM)	UA (Btu/hr ft F)	T pipe (F)	T air (F)	TEMP. DIFF. (F)
0	0.1335	113.9	71.8	42.1
0	0.1353	115.7	74.7	41.0
0	0.1364	115.0	74.4	40.6
0	0.1398	115.0	73.5	41.6
0.48	0.1417	116.1	71.2	44.9
2.08	0.1642	116.2	72.5	43.7
2.96	0.1837	115.6	73.6	42.0
4.17	0.2082	117.3	74.5	42.8
0	0.1302	112.7	63.8	48.9
0	0.1318	114.4	68.3	46.2
0	0.1341	114.3	66.1	48.2
1.15	0.1609	116.5	69.5	47.0
3.63	0.1807	115.6	66.8	48.7
0	0.1278	114.5	61.0	53.5
3.12	0.1695	115.8	65.0	50.8
4.12	0.1849	114.3	62.4	51.9
0	0.1233	112.7	56.6	56.2
0	0.1235	115.5	56.4	59.1
0	0.1236	114.1	56.5	57.6
0	0.1236	114.5	55.8	58.7
0	0.1240	114.2	56.2	58.0
0	0.1251	114.9	56.8	58.1
0	0.1265	112.8	54.9	57.9
0	0.1289	114.5	58.6	56.0
0	0.1323	114.5	56.9	57.6
0.50	0.1449	116.1	56.8	59.3
1.19	0.1605	116.6	56.6	60.0
2.09	0.1655	116.2	57.6	58.6
2.94	0.1646	115.7	59.5	56.2
3.15	0.1788	116.6	56.6	60.0
0	0.1274	130.8	67.8	63.0
0	0.1297	131.4	69.0	62.5
0.47	0.1391	116.1	54.6	61.4
1.13	0.1505	116.6	56.1	60.4
2.06	0.1729	116.6	56.5	60.1
2.90	0.1923	116.6	56.2	60.4
3.79	0.2016	116.5	56.3	60.2

0	0.1231	130.3	60.5	69.8
0	0.1255	132.8	66.3	66.5
0	0.1288	132.4	63.3	69.2
1.17	0.1650	134.8	69.0	65.8
2.04	0.1794	134.2	68.7	65.5
2.96	0.1789	133.9	67.2	66.7
0	0.1245	131.4	57.9	73.5
0	0.1251	133.7	58.9	74.8
0	0.1260	128.7	56.3	72.4
0	0.1264	131.5	57.1	74.5
1.16	0.1616	136.2	62.3	73.9
4.22	0.1950	136.6	64.4	72.2
0	0.1255	132.1	56.5	75.7
0.46	0.1436	133.4	56.5	77.0
0.48	0.1481	135.1	59.7	75.4
1.16	0.1580	135.1	57.6	77.6
2.08	0.1647	134.4	56.6	77.8
2.94	0.1804	136.3	58.2	78.1
3.02	0.1816	134.5	56.4	78.2
4.17	0.1861	132.8	55.8	76.9

Table D-4
UA Summary Table – 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation

FLOW RATE (GPM)	UA (Btu/hr ft F)	T pipe (F)	T air (F)	TEMP. DIFF. (F)
0	0.1143	115.8	70.6	45.2
0	0.1163	114.6	67.2	47.4
0.47	0.1334	116.7	73.2	43.5
3.02	0.1623	115.6	65.9	49.8
3.05	0.1819	117.0	68.4	48.5
0	0.1129	115.3	64.5	50.9
0	0.1236	114.9	62.2	52.7
1.58	0.1516	116.2	63.0	53.2
1.61	0.1527	115.9	61.4	54.5
0	0.1102	112.8	56.2	56.6
0	0.1112	115.2	60.0	55.1
0	0.1126	130.6	70.7	59.9
0	0.1127	114.5	56.8	57.6
0	0.1163	112.1	56.7	55.4
0	0.1177	115.1	58.3	56.8
0	0.1254	113.1	56.4	56.7
0.44	0.1286	115.6	56.6	59.0
0.45	0.1257	116.0	56.0	60.0
0.47	0.1391	115.3	56.5	58.8
1.07	0.1410	116.3	57.7	58.6
1.13	0.1459	116.3	59.1	57.2
4.04	0.1844	116.1	56.3	59.8
0	0.1121	131.9	69.5	62.4
0.45	0.1324	134.1	71.2	62.9
0	0.1113	133.8	67.2	66.6
0	0.1114	131.9	66.6	65.3
0	0.1133	132.0	65.2	66.8
0	0.1142	130.7	61.0	69.7
0	0.1147	133.8	68.6	65.2
0	0.1162	128.1	58.9	69.2
1.11	0.1502	133.4	67.2	66.2
1.97	0.1570	133.8	67.7	66.1
2.99	0.1826	134.6	65.8	68.8
3.40	0.1811	133.3	65.8	67.5
4.07	0.1754	135.0	69.6	65.4
4.19	0.1649	134.9	65.5	69.3

0	0.1126	133.2	59.4	73.8
0	0.1139	133.8	61.9	71.9
0	0.1147	134.8	64.7	70.1
0	0.1152	134.0	59.6	74.4
0	0.1170	133.5	60.2	73.3
0	0.1171	134.0	59.4	74.6
0	0.1233	131.1	59.8	71.3
0	0.1282	134.2	63.0	71.1
0.43	0.1312	134.5	59.9	74.6
0.45	0.1316	131.4	58.6	72.8
0.46	0.1296	134.2	60.4	73.9
1.17	0.1469	135.6	61.5	74.1
1.51	0.1507	136.0	62.4	73.7
2.98	0.1831	136.1	63.4	72.7
1.12	0.1519	135.7	59.6	76.1
1.89	0.1692	136.1	59.4	76.8
1.93	0.1710	136.0	59.4	76.5
2.70	0.1872	135.2	59.5	75.8
3.01	0.1957	135.2	60.0	75.2
3.42	0.1730	136.4	60.0	76.3

Table D-5
UA Summary Table – 1/2 Inch Rigid Copper Pipe Curve Fits to High Data

FLOW RATE (GPM)	UA BARE (BTU/HR FT F)	UA R-3.1 (BTU/HR FT F)	UA R-5.2 (BTU/HR FT F)
0	0.226	0.128	0.116
0.5	0.33	0.145	0.135
1.0	0.345	0.155	0.145
1.5	0.35	0.168	0.157
2.0	0.36	0.18	0.17
3.0	0.36	0.195	0.188
4.0	0.36	0.2	0.19
5.0	0.36	0.2	0.19

Table D-6
AF/PV Summary Table – 1/2 Inch Bare Rigid Copper Pipe

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.1 FT. (0.256 gal.)	AF/PV AT L=42.6 FT. (0.5168 gal.)	AF/PV AT L=64.2 FT. (0.7779 gal.)	AF/PV AT L=85.7 FT. (1.0387 gal.)	AF/PV AT L=107.3 FT. (1.2999 gal.)	AF/PV AT L=128.2 FT. (1.5531 gal.)
0.37	115.5	60.0	60.0	0.189	1.628	1.527	1.584	1.512	1.551	1.752
0.45	115.9	60.0	71.4	0.195	1.665	1.454	1.497	1.442	1.448	1.465
0.98	116.0	59.8	72.8	0.196	1.497	1.402	1.426	1.401	1.387	1.398
0.43	115.3	69.0	60.0	0.222	1.615	1.494	1.540	1.488	1.509	1.578
0.45	116.6	69.5	69.5	0.246	1.599	1.453	1.473	1.432	1.440	1.448
1.02	115.6	69.0	60.0	0.227	1.478	1.392	1.424	1.397	1.391	1.396
1.05	116.1	71.7	70.2	0.250	1.441	1.382	1.392	1.375	1.384	1.397
1.09	115.6	73.2	73.8	0.250	1.464	1.387	1.413	1.386	1.385	1.398
1.80	115.8	69.8	60.0	0.235	1.354	1.306	1.381	1.341	1.338	1.368
3.03	116.0	68.0	60.0	0.229	1.461	1.281	1.421	1.340	1.348	1.348
3.74	116.1	66.5	60.0	0.224	1.703	1.486	1.423	1.367	1.345	1.325
0.44	115.8	74.8	82.0	0.263	1.460	1.400	1.425	1.388	1.396	1.394
0.44	116.4	74.8	69.7	0.274	1.481	1.387	1.441	1.393	1.408	1.417
0.45	114.1	80.8	83.0	0.273	1.406	1.363	1.416	1.377	1.372	1.383
0.46	116.7	74.0	83.5	0.274	1.470	1.421	1.440	1.388	1.404	1.464
0.46	117.2	75.8	78.6	0.295	1.466	1.366	1.373	1.368	1.359	1.368
0.47	115.8	75.5	80.7	0.268	1.460	1.325	1.394	1.373	1.372	1.385
0.98	115.8	75.0	82.0	0.265	1.428	1.358	1.384	1.358	1.357	1.354
1.00	116.5	74.7	84.0	0.275	1.375	1.374	1.378	1.365	1.361	1.356
1.72	116.4	71.3	69.9	0.253	1.438	1.324	1.378	1.349	1.352	1.362
1.83	116.2	75.1	84.0	0.273	1.432	1.329	1.334	1.335	1.339	1.331
2.09	115.8	75.5	75.8	0.268	1.294	1.283	1.345	1.329	1.330	1.349
2.14	117.1	73.3	73.7	0.276	1.458	1.363	1.380	1.345	1.347	1.355
2.83	116.3	72.3	69.8	0.257	1.285	1.428	1.330	1.360	1.332	1.350
3.02	115.9	75.2	83.5	0.268	1.293	1.404	1.327	1.349	1.335	1.307
4.12	115.8	75.0	83.0	0.265	1.442	1.352	1.352	1.334	1.330	1.321
0.47	121.7	72.8	69.0	0.342	1.391	1.358	1.409	1.353	1.375	1.375
1.02	121.5	71.6	69.4	0.331	1.326	1.330	1.362	1.342	1.336	
1.54	120.7	71.8	69.2	0.321	1.307	1.335	1.356	1.322	1.324	1.327
3.10	119.8	72.3	69.4	0.312	1.482	1.299	1.326	1.316	1.302	1.280
0.42	115.4	88.0	60.5	0.380	1.350	1.363	1.422	1.406	1.437	1.481
1.02	115.8	88.6	62.0	0.397	1.317	1.311	1.359	1.335	1.351	1.343
0.44	130.4	72.6	72.4	0.439	1.348	1.308	1.339	1.418	1.339	1.347
0.46	135.3	65.0	65.4	0.431	1.289	1.312	1.355	1.338	1.345	1.346

0.49	133.5	69.5	68.0	0.445	1.356	1.322	1.436	1.426	1.330	1.349
1.60	135.7	66.7	65.5	0.445	1.252	1.244	1.280	1.276	1.293	1.283
1.85	116.0	89.5	62.0	0.415	1.318	1.293	1.305	1.302	1.309	1.307
3.20	116.0	89.7	63.0	0.418	1.284	1.182	1.260	1.294	1.278	1.266
0.45	134.8	70.1	67.8	0.461	1.316	1.315	1.347	1.407	1.323	1.338
0.48	133.9	72.8	68.9	0.473	1.281	1.303	1.396	1.385	1.317	1.325
0.94	136.4	69.3	65.5	0.468	1.257	1.276	1.283	1.290	1.304	1.305
1.62	136.3	68.8	65.6	0.463	1.259	1.231		1.265	1.288	1.275
3.02	136.7	67.7	65.1	0.459	1.167	1.109	1.250	1.234	1.237	1.241
4.04	135.8	69.7	65.3	0.466	1.000	1.098	1.165	1.192	1.220	1.209
0.46	117.4	93.6	77.5	0.521	1.252	1.275	1.292	1.301	1.316	1.338
0.47	129.8	96.2	99.0	0.738	1.156	1.181	1.236	1.231	1.237	1.261

Table D-7
AF/PV Summary Table – 1/2 Inch Rigid Copper Pipe With R-3.1 Insulation

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.1 FT. (0.256 gal.)	AF/PV AT L=42.6 FT. (0.5168 gal.)	AF/PV AT L=64.2 FT. (0.7779 gal.)	AF/PV AT L=85.7 FT. (1.0387 gal.)	AF/PV AT L=107.3 FT. (1.2999 gal.)	AF/PV AT L=128.2 FT. (1.5531 gal.)
0.46	116.2	55.6	54.5	0.185	1.605	1.543	1.506	1.490	1.482	1.482
1.13	116.7	56.0	55.9	0.193	1.511	1.474	1.452	1.438	1.423	1.419
2.98	115.8	59.5	59.4	0.192	1.695	1.374	1.439	1.370	1.395	1.375
3.16	116.7	57.0	56.6	0.196	1.611	1.342	1.441	1.392	1.349	1.375
3.62	115.7	62.0	67.0	0.199	1.473	1.342	1.323	1.398	1.395	1.378
0.48	116.2	63.3	70.6	0.212	1.523	1.440	1.461	1.440	1.432	1.444
1.16	116.6	62.8	69.2	0.215	1.438	1.421	1.410	1.382	1.391	1.393
2.14	116.3	60.4	57.5	0.202	1.548	1.419	1.461	1.364	1.367	1.357
2.14	116.3	62.9	72.5	0.211	1.433	1.396	1.380	1.376	1.373	1.372
2.98	116.3	62.5	73.7	0.210	1.303	1.442	1.375	1.366	1.374	1.344
3.10	115.9	61.4	65.0	0.200	1.627	1.430	1.331	1.395	1.365	1.335
4.12	117.3	62.9	74.7	0.226	1.554	1.431	1.379	1.363	1.356	1.345
0.45	133.5	57.8	56.1	0.377	1.396	1.373	1.369	1.379	1.377	1.380
0.50	116.2	87.2	56.7	0.386	1.327	1.353	1.376	1.365	1.385	
2.13	116.7	87.3	56.4	0.397	1.305	1.284	1.279	1.298	1.303	1.303
2.16	134.6	58.6	56.5	0.389	1.166	1.247	1.284	1.298	1.308	1.308
2.90	116.7	87.2	56.1	0.396	1.363	1.214	1.273	1.280	1.267	1.285
3.00	134.7	59.9	56.4	0.397	1.171	1.260	1.303	1.266	1.304	1.292
4.22	132.9	56.9	55.8	0.367	1.304	1.184	1.239	1.231	1.244	1.241
1.17	135.3	59.9	57.4	0.402	1.318	1.344	1.323	1.332	1.336	1.340
1.29	116.7	87.6	56.6	0.401	1.272	1.298	1.310	1.322	1.333	1.323
3.00	136.5	59.8	58.2	0.411	1.160	1.185	1.296	1.260	1.287	1.280
3.80	116.7	87.9	56.2	0.406	1.005	1.140	1.207	1.243	1.272	1.251
0.48	135.2	87.2	59.6	0.629	1.299	1.300	1.298	1.306	1.319	1.307
1.17	136.3	89.2	62.3	0.664	1.093	1.201	1.227	1.237	1.251	1.264
1.18	135.0	88.6	68.9	0.647	1.121	1.191	1.218	1.238	1.254	1.258
2.13	134.4	88.9	68.6	0.646	1.000	1.126	1.174	1.212	1.225	1.238
2.95	134.1	88.6	67.2	0.640	1.000	1.008	1.195	1.188	1.195	1.221
4.23	136.8	87.9	64.8	0.650	1.063	1.136	1.147	1.132	1.146	1.143

Table D-8
AF/PV Summary Table – 1/2 Inch Rigid Copper Pipe With R-5.2 Insulation

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=21.1 FT. (0.256 gal.)	AF/PV AT L=42.6 FT. (0.5168 gal.)	AF/PV AT L=64.2 FT. (0.7779 gal.)	AF/PV AT L=85.7 FT. (1.0387 gal.)	AF/PV AT L=107.3 FT. (1.2999 gal.)	AF/PV AT L=128.2 FT. (1.5531 gal.)
0.47	115.4	56.7	56.13	0.177	1.645	1.504	1.506	1.481	1.490	1.492
0.45	115.7	65.8	56.4	0.215	1.526	1.460	1.458	1.457	1.471	1.468
1.12	116.4	65.6	59.1	0.224	1.397	1.409	1.410	1.407	1.399	1.397
1.61	116.3	66.9	63.0	0.229	1.479	1.410	1.390	1.382	1.371	1.368
3.04	117.0	67.2	68.6	0.241	1.242	1.444	1.343	1.360	1.369	1.331
4.02	116.5	64.8	56.2	0.222	1.306	1.327	1.320	1.318	1.323	1.312
0.46	116.1	82.3	55.7	0.328	1.395	1.393	1.408	1.408	1.399	1.411
0.46	131.4	61.1	58.1	0.376	1.357	1.373	1.362	1.354	1.365	1.365
1.58	116.0	88.2	61.4	0.395	1.275	1.297	1.323	1.311	1.324	1.326
0.43	134.6	63.2	59.4	0.414	1.325	1.337	1.355	1.355	1.353	1.353
0.46	134.4	68.7	60.0	0.448	1.361	1.361	1.343	1.343	1.348	1.348
0.47	116.7	89.6	73.0	0.431	1.304	1.321	1.335	1.336	1.347	1.360
1.07	116.4	90.0	57.5	0.431	1.296	1.305	1.333	1.322	1.336	1.331
1.12	135.8	66.2	59.3	0.443	1.204	1.286	1.309	1.305	1.307	1.315
1.98	136.1	65.6	59.2	0.441	1.313	1.293	1.295	1.292	1.296	1.302
2.58	135.3	64.4	59.5	0.427	1.250	1.262	1.251	1.285	1.272	1.283
3.04	115.9	89.7	66.0	0.416	1.018	1.309	1.279	1.251	1.304	1.290
3.47	136.5	60.7	60.0	0.416	1.095	1.148	1.201	1.279	1.282	1.271
1.18	135.7	67.9	61.5	0.453	1.268	1.273	1.301	1.308	1.319	1.314
1.61	136.2	67.5	62.3	0.454	1.251	1.246	1.292	1.285	1.298	1.294
3.04	136.2	69.7	63.5	0.469	1.275	1.150	1.261	1.265	1.261	1.255
4.40	135.3	70.4	65.6	0.467	1.108	1.244	1.263	1.252	1.263	1.217
1.13	133.5	86.4	67.1	0.605	1.114	1.201	1.248	1.269	1.271	1.281
4.29	135.4	86.0	69.7	0.615		1.023	1.051	1.091	1.141	1.168
0.45	134.2	92.8	70.9	0.705	1.186	1.214	1.241	1.255	1.230	1.278
1.96	136.3	90.2	59.2	0.678	1.000	1.142	1.192	1.202	1.226	1.231
2.01	134.0	91.7	67.7	0.685	1.050	1.160	1.210	1.206	1.237	1.223
3.01	134.9	91.2	65.7	0.683	1.075	1.081	1.109	1.202	1.201	1.197

E
APPENDIX – 3/4 INCH PAX PIPE TESTS - DETAILED DATA

Technical information on the 3/4 inch PAX piping system was as follows:

Table E-1
3/4 Inch PAX Piping Technical Data

	WATER	PIPE	½ FOAM (R-3.1)	¾ FOAM (R-5.2)
ID (inches)	-	0.806	0.875**	0.875**
OD (inches)	0.806	1.000	1.875	2.375
Density(lbm/ft^3)	62.0*	559	0.58	0.58
Cp (Btu/lbm F)	1.0	0.0915	0.48	0.48
mCp (Btu/ft F)	0.2211	0.0617	0.0042	0.0074

*Water density at 100 F

** Note interference fit – insulation was stretched over the pipe

Figure E-1
¾ Inch Bare PAX Test Section

Figure E-2
¾ Inch Insulated PAX Test Section

Table E-2
UA Summary Table – 3/4 Inch Bare PAX Pipe

FLOW RATE (GPM)	UA (Btu/hr ft F)	T pipe (F)	T air (F)	TEMP. DIFF. (F)
0	0.526			
0	0.507			
0	0.522			
0	0.512			
0	0.527			
0	0.522			
0	0.514			
0	0.499			
0	0.496			
0	0.523			
0	0.545			
0	0.526			
0	0.517			
0	0.488			
0	0.527			
0	0.489			
0	0.486			
0	0.494			
0	0.512			
0	0.494			
0	0.517			
0	0.557			
0	0.521			
0	0.521			
0	0.53			
0	0.518			
0	0.501			
0	0.491			
0	0.532			
0	0.517			
0	0.474			
0	0.501			
0	0.461			
0	0.519			
0	0.525			
0	0.487			
0	0.524			
0	0.527			
0	0.525			
0	0.529			
0	0.523			

0	0.506			
0	0.496			
0	0.501			
0	0.494			
0	0.486			
0	0.486			
0	0.493			
0	0.481			
0	0.476			
0	0.486			
0	0.486			
0	0.522			
0.47	0.526	117.5	58.6	58.9
0.48	0.521	132.2	71.5	60.7
0.48	0.509	115.6	52.1	63.6
0.48	0.502	115.7	53.7	62.1
0.50	0.523	115.8	49.3	66.5
0.90	0.517	115.9	53.0	63.0
1.02	0.507	116.4	57.4	59.0
1.81	0.493	116.7	56.9	59.8
3.05	0.467	116.6	56.2	60.4
3.13	0.473	116.4	55.7	60.6
4.46	0.426	116.3	60.0	56.3
0.48	0.528	132.1	63.0	69.1
0.90	0.504	115.9	51.5	64.4
0.90	0.496	116.5	53.1	63.4
0.91	0.512	116.1	47.6	68.5
0.92	0.521	132.7	66.9	65.8
0.94	0.528	133.4	69.7	63.6
0.94	0.509	116.1	51.7	64.4
1.77	0.507	133.8	69.5	64.3
1.79	0.482	116.6	53.3	63.3
1.83	0.494	116.4	53.5	62.9
1.83	0.501	116.4	52.8	63.6
1.87	0.496	116.1	51.7	64.3
1.88	0.506	116.3	52.0	64.3
1.88	0.483	116.0	52.9	63.2
2.70	0.471	116.2	52.7	63.4
2.99	0.489	116.5	50.8	65.7
3.07	0.484	116.7	53.3	63.4
3.23	0.485	134.4	70.8	63.6
3.84	0.466	116.6	53.7	62.9
4.31	0.460	116.4	50.9	65.5
4.32	0.493	116.2	51.7	64.5

4.37	0.442	116.9	52.6	64.3
4.39	0.376	115.6	52.0	63.6
4.40	0.480	116.7	52.2	64.5
4.54	0.483	134.2	70.3	63.9
4.55	0.479	116.5	51.9	64.7
0.46	0.531	132.9	58.8	74.2
0.47	0.531	133.6	59.0	74.7
0.47	0.531	132.7	58.4	74.4
0.47	0.540	133.8	59.4	74.4
0.50	0.505	123.7	49.4	74.3
0.90	0.520	133.5	59.4	74.0
0.91	0.524	131.1	58.2	72.9
1.81	0.529	135.1	65.7	69.4
2.59	0.519	135.0	66.4	68.6
2.90	0.517	116.8	50.2	66.6
3.22	0.491	134.4	64.8	69.6
4.43	0.463	116.6	48.4	68.2
4.46	0.475	116.4	49.2	67.3
4.47	0.480	127.7	58.2	69.4
0.46	0.526	131.5	50.8	80.6
0.52	0.546	133.7	53.2	80.5
0.53	0.530	131.3	52.0	79.3
0.53	0.545	132.4	50.3	82.1
0.87	0.495	125.1	48.7	76.4
0.89	0.530	132.8	54.0	78.8
0.89	0.525	133.0	54.3	78.7
0.95	0.537	134.1	56.5	77.6
1.77	0.532	131.1	54.0	77.1
1.82	0.528	132.9	58.9	74.0
1.84	0.528	133.7	58.2	75.6
2.02	0.521	133.4	57.2	76.2
3.73	0.485	133.5	59.1	74.4
4.26	0.482	134.0	61.3	72.7
4.61	0.469	134.8	60.6	74.2
0.49	0.522	133.3	47.3	86.0
0.89	0.528	133.0	48.2	84.8
0.92	0.519	132.6	49.4	83.3
0.95	0.529	134.7	52.0	82.7
0.98	0.537	132.3	50.9	81.3
1.70	0.528	133.1	52.1	81.0
2.83	0.453	126.4	47.5	78.8
3.27	0.530	135.3	57.7	77.6
3.43	0.478	131.9	51.9	80.0
3.74	0.500	134.8	58.8	76.0

1.72	0.521	135.4	52.5	82.9
1.80	0.510	134.6	50.1	84.5
1.90	0.523	133.5	50.6	82.9
2.13	0.519	132.7	50.5	82.2
2.67	0.502	133.6	48.2	85.4
2.83	0.496	135.2	53.3	81.9
3.08	0.505	134.0	50.3	83.7
3.62	0.520	133.6	51.3	82.3
3.99	0.472	134.6	50.5	84.1
4.34	0.491	132.6	51.2	81.4
4.37	0.476	133.5	48.6	84.9
4.55	0.481	134.0	50.5	83.6
4.82	0.506	132.8	51.9	80.9
4.84	0.504	134.3	53.3	81.1
3.84	0.458	135.8	46.6	89.2

Table E-3
UA Summary Table – 3/4 Inch PAX Pipe With R-4.7 Insulation

FLOW RATE (GPM)	UA (Btu/hr ft F)	T pipe (F)	T air (F)	TEMP. DIFF. (F)
0	0.157			
0	0.154			
0	0.156			
0	0.162			
0	0.156			
0	0.154			
0	0.156			
0	0.155			
0	0.162			
0	0.16			
0	0.17			
0	0.158			
0	0.156			
0	0.164			
0	0.158			
0	0.157			
0	0.153			
0	0.156			
0	0.158			
0	0.16			
0	0.155			
0	0.159			
0	0.153			
0	0.163			
4.46	0.102	116.5	67.1	49.4
4.48	0.124	116.9	67.0	49.9
0.46	0.167	116.8	61.9	54.9
0.48	0.159	116.0	64.3	51.7
0.97	0.159	116.5	65.2	51.3
1.84	0.148	116.9	66.1	50.8
3.12	0.130	116.8	63.3	53.5
4.53	0.106	116.8	59.9	56.9
0.46	0.155	115.4	48.3	67.0
0.49	0.157	115.5	50.2	65.4
0.92	0.155	116.3	50.4	65.9
1.77	0.143	116.4	54.2	62.2
0.94	0.166	116.3	49.1	67.2

1.89	0.142	116.5	48.7	67.9
3.20	0.135	116.7	49.6	67.1
3.21	0.160	116.6	48.4	68.3
4.57	0.128	116.7	48.4	68.3
4.58	0.117	116.1	50.3	65.8
0.48	0.168	133.7	58.1	75.6
0.92	0.167	133.3	60.3	73.1
2.02	0.163	135.1	60.3	74.8
3.12	0.153	134.8	60.1	74.7
0.45	0.157	132.3	49.8	82.5
0.48	0.164	134.9	54.1	80.8
0.48	0.167	133.0	50.2	82.8
0.50	0.164	132.5	54.2	78.3
0.53	0.169	129.6	51.0	78.6
0.83	0.157	134.5	53.7	80.8
0.92	0.178	134.7	53.7	81.0
3.12	0.158	133.8	54.5	79.3
0.91	0.163	135.2	50.4	84.7
1.58	0.193	136.7	54.6	82.1
1.85	0.176	135.1	53.4	81.7
1.97	0.166	135.2	53.6	81.6
2.03	0.154	131.2	49.8	81.4
3.17	0.198	136.8	55.3	81.5
3.18	0.152	133.1	51.9	81.2
3.30	0.161	135.8	53.4	82.4
3.44	0.139	133.2	50.8	82.4
3.51	0.170	133.7	51.1	82.6
3.65	0.147	136.6	54.3	82.3
4.86	0.132	133.9	53.2	80.7
4.91	0.144	133.0	51.0	82.0
1.84	0.147	135.3	49.6	85.7
3.30	0.153	135.4	49.6	85.7
4.60	0.161	136.2	50.5	85.6

Table E-4
UA Summary Table – 3/4 Inch PAX Pipe Curve Fits to High Data

FLOW RATE (GPM)	UA BARE (BTU/HR FT F)	UA R-4.7 (BTU/HR FT F)
0	0.55	0.158
0.5	0.546	0.167
1.0	0.541	0.180
1.5	0.537	0.177
2.0	0.532	0.175
3.0	0.522	0.174
4.0	0.513	0.17
5.0	0.503	0.16

Table E-5
AF/PV Summary Table – 3/4 Inch Bare PAX Pipe

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=25.3 FT. (0.6697 gal.)	AF/PV AT L=50.3 FT. (1.3328 gal.)	AF/PV AT L=75.5 FT. (2.0012 gal.)	AF/PV AT L=100.2 FT. (2.6557 gal.)
0.47	115.9	56.2	53.5	0.182	1.632	1.645	1.743	999.000
0.47	115.4	55.8	52.1	0.175	1.639	1.702	1.826	999.000
0.90	116.0	55.8	52.5	0.183	1.452	1.504	1.498	1.512
0.91	116.5	56.0	52.9	0.190	1.497	1.493	1.494	1.504
0.92	116.1	48.1	46.5	0.163	1.564	1.587	1.597	1.595
1.02	116.5	56.9	57.3	0.193	1.406	1.470	1.474	1.466
1.83	116.6	56.4	52.9	0.193	1.277		1.439	1.431
1.90	116.7	56.9	56.7	0.196	1.250	1.410	1.426	1.421
1.91	116.5	56.6	52.9	0.192	1.226	1.409	1.433	1.432
1.95	116.2	54.2	50.9	0.181	1.260	1.446	1.458	1.448
1.97	116.2	56.1	52.2	0.186	1.251	1.424	1.452	1.438
2.81	116.1	56.6	52.3	0.187	1.103	1.319	1.383	1.394
3.11	116.4	56.5	55.5	0.190	1.149	1.306	1.380	1.390
3.12	116.7	55.5	52.7	0.191	1.073	1.303	1.378	1.392
3.91	116.5	56.4	53.1	0.191	1.000	1.183	1.326	1.344
4.40	115.5	55.5	51.2	0.175	1.176	1.192	1.320	1.355
4.47	115.8	51.0	49.8	0.167	1.225	1.276	1.406	1.424
4.50	116.4	56.6	59.9	0.191	1.035	1.154	1.278	1.314
0.47	117.3	55.9	58.3	0.200	1.585	1.592	1.650	1.846
3.06	117.6	56.3	57.7	0.206	1.000			1.340
0.50	124.0	54.3	48.8	0.272	1.410	1.439	1.449	1.467
0.89	125.4	54.3	47.9	0.287	1.244	1.334	1.342	1.346
2.84	126.5	46.0	46.5	0.267	1.000	1.131	1.215	1.237
0.45	131.6	54.8	50.6	0.346	1.338	1.367	1.357	1.367
0.90	132.8	47.1	47.0	0.325	1.165	1.268	1.274	1.286
1.88	131.1	53.5	53.0	0.337	1.000	1.130	1.195	1.212
4.39	132.7	52.3	50.3	0.344	1.000	1.016	1.044	1.057
0.46	133.0	56.3	56.9	0.365	1.301	1.308	1.325	1.349
0.47	132.3	58.0	70.3	0.368	1.253	1.286	1.289	1.300
0.48	133.8	55.0	52.7	0.365	1.423	1.377	1.366	1.371
0.51	134.2	55.1	46.9	0.369	1.333	1.323	1.346	1.354
0.92	133.1	56.1	53.3	0.365	1.098	1.209	1.241	1.252
0.93	132.8	54.1	48.7	0.353	1.120	1.220	1.263	1.272
0.95	133.4	57.1	69.2	0.372	1.089	1.203	1.227	1.230
0.97	134.8	55.1	51.2	0.374	1.135	1.217	1.244	1.254
1.83	135.5	55.4	51.4	0.381	1.039	1.091	1.150	1.178

1.91	134.6	54.9	49.2	0.371	1.044	1.088	1.155	1.182
1.96	135.1	56.7	65.1	0.384	1.000	1.078	1.138	1.164
2.25	132.9	57.7	49.9	0.371	1.056	1.087	1.148	1.175
2.71	133.7	53.7	47.1	0.359	1.000	1.024	1.103	1.134
2.83	135.4	56.7	52.2	0.386	1.000	1.016	1.045	1.091
2.94	135.0	57.7	64.0	0.388	1.000	1.025	1.059	1.082
3.27	134.5	57.9	70.2	0.385	1.002	1.001	1.051	1.069
3.93	135.8	51.4	45.3	0.365	1.000	1.000	1.014	1.053
3.95	134.7	55.0	49.7	0.373	1.000	1.000	1.022	1.043
4.38	134.0	56.4	59.0	0.373	1.032	1.000	1.000	1.007
4.38	116.7	87.0	47.7	0.393	1.057	1.000	1.022	1.038
4.63	134.3	58.0	69.2	0.384	1.093	1.000	1.000	1.026
4.92	134.5	56.5	52.1	0.378	1.000	1.014	1.000	1.027
0.47	115.8	89.9	51.9	0.417	1.194	1.382	1.551	999.000
0.48	115.9	90.1	48.9	0.422	1.198	1.380	1.664	999.000
1.90	116.3	90.9	51.8	0.445	1.002	1.047	1.136	1.173
3.17	115.9	90.6	52.7	0.430	1.022	1.000	1.036	1.086
4.44	116.9	90.2	52.8	0.445	1.000	1.000	1.011	0.999
0.90	116.0	91.8	51.1	0.454	1.020	1.189	1.266	1.333
0.94	116.1	91.6	51.4	0.453	1.020	1.180	1.248	1.312
1.83	116.5	92.2	52.5	0.472	1.000	1.063	1.134	1.172
3.07	116.7	91.9	56.4	0.472	1.000	1.006	1.020	1.034
3.15	116.9	92.3	49.5	0.484	1.000	1.000	1.017	1.043
4.44	116.8	92.6	52.0	0.487	1	1	1.006	1
4.46	116.5	92.9	48.8	0.486	1.012	1	1	1
3.04	116.6	94.3	50.2	0.519	1	1	1.003	1.020
0.52	132.4	86.6	50.2	0.598	1	1.129	1.187	1.220
0.52	131.4	90.6	51.7	0.647	1	1.096	1.176	1.206
1.97	133.1	86.8	51.3	0.606	1	1	1.013	1.028
1.99	133.7	88.4	50.3	0.633	1	1	1	1.018
0.47	132.2	93.1	62.4	0.695	1.003	1.105	1.148	1.196
0.92	133.1	90.7	53.9	0.663	1.031	1.018	1.072	1.115
0.97	132.4	90.9	50.6	0.660	1	1.011	1.077	1.113
1.05	134.2	92.0	56.0	0.691	1	1.005	1.054	1.087
3.12	134.1	90.7	49.3	0.670	1	1	1	1
3.55	132.1	92.7	51.5	0.688	1	1	1	1
3.62	133.9	91.7	50.9	0.684	1	1	1	1
4.66	134.1	91.4	50.1	0.681	1	1	1	1
4.78	132.8	90.3	51.3	0.653	1	1	1	1
0.47	133.8	94.0	58.1	0.723	1	1.064	1.118	1.179

0.93	132.8	94.0	66.3	0.716	1	1	1.046	1.074
1.87	133.8	93.3	68.8	0.711	1	1	1	1.005
3.20	134.6	94.1	64.4	0.731	1	1	1	1
4.69	135.1	92.3	60.2	0.703	1	1	1	1

Table E-6
AF/PV Summary Table – 3/4 Inch PAX Pipe With R-4.7 Insulation

FLOW RATE (GPM)	T hot (F)	T pipe initial (F)	Tair (F)	TDR	AF/PV AT L=25.3 FT. (0.6697 gal.)	AF/PV AT L=50.3 FT. (1.3328 gal.)	AF/PV AT L=75.5 FT. (2.0012 gal.)	AF/PV AT L=100.2 FT. (2.6557 gal.)
0.45	116.3	57.4	62.4	0.192	1.572	1.547	1.489	1.486
0.48	116.9	56.8	61.6	0.198	1.569	1.538	1.499	1.492
0.50	115.7	53.8	50.0	0.173	1.687	1.595	1.578	1.555
0.97	116.6	57.6	64.5	0.197	1.436	1.467	1.454	1.426
0.97	116.3	53.9	49.0	0.181	1.504	1.528	1.502	1.470
1.94	116.6	54.2	48.5	0.186	1.283	1.427	1.463	1.441
3.20	116.9	54.1	49.4	0.189	1.074	1.300	1.388	1.384
4.46	117.0	56.7	59.7	0.199	1.155	1.155	1.277	1.299
4.67	116.2	53.1	49.9	0.178	1.052	1.188	1.326	1.367
1.90	116.9	58.0	65.6	0.202		1.369	1.407	1.436
3.15	117.0	57.6	63.0	0.202	1.060	1.264	1.361	1.370
4.55	117.0	57.9	66.8	0.203	1.000	1.112	1.245	1.293
0.47	132.6	52.7	54.0	0.345	1.341	1.317	1.316	1.313
0.47	132.6	52.5	49.5	0.344	1.342	1.328	1.333	1.330
0.53	129.6	59.2	50.4	0.349	1.242	1.310	1.299	1.304
0.48	133.9	52.5	57.1	0.355	1.365	1.344	1.329	1.339
0.89	134.0	52.8	58.2	0.357	1.116	1.219	1.252	1.255
0.89	134.6	52.8	53.4	0.362	1.078	1.206	1.248	1.248
0.89	135.3	52.2	49.8	0.365	1.099	1.211	1.246	1.244
0.92	133.6	53.0	60.1	0.355	1.104	1.222	1.250	1.248
2.07	135.5	51.4	49.3	0.363	1.000	1.071	1.140	1.170
2.09	135.3	52.7	53.4	0.367	1.000	1.060	1.134	1.160
2.18	135.1	53.7	59.4	0.370	1.022	1.064	1.123	1.156
3.16	135.0	53.8	59.9	0.369	1.000	1.021	1.049	1.087
3.32	135.6	52.2	49.2	0.367	1.000	1.028	1.052	1.078
3.36	135.9	52.8	53.3	0.372	1.000	1.000	1.045	1.062
3.57	133.7	52.7	50.8	0.354	1.000	1.029	1.039	1.079
3.66	136.5	49.6	53.4	0.363	1.000	1.000	1.015	1.052
4.65	136.3	52.2	50.0	0.372	1.000	1.000	1.000	1.000
4.95	134.6	53.0	52.8	0.363	1.000	1.000	1.000	1.000
0.48	116.2	90.4	65.2	0.433	1.138	1.233	1.267	1.283
0.48	115.5	91.4	48.2	0.435	1.168	1.275	1.319	1.326
2.11	131.2	67.1	49.4	0.408	1.000	1.044	1.103	1.132
3.12	133.7	64.8	54.2	0.417	1.000	1.000	1.012	1.034
0.92	116.3	91.4	49.2	0.454	1.040	1.154	1.200	1.223

1.83	116.4	91.5	53.9	0.458	1.000	1.048	1.105	1.138
3.23	116.6	91.3	48.2	0.458	1.000	1.000	1.024	1.024
4.49	116.5	91.3	67.2	0.455	1.000	1.003	1.000	1.000
4.60	116.8	91.8	48.3	0.472	1.000	1.000	1.000	1.000
0.48	133.2	88.6	49.5	0.632	1.000	1.075	1.124	1.145
0.49	135.1	88.6	53.7	0.647	1.046	1.114	1.138	1.160
0.82	132.5	88.6	52.8	0.626	1.000	1.017	1.080	1.105
0.91	134.7	89.2	53.6	0.652	1.000	1.000	1.050	1.078
1.65	136.8	89.0	54.3	0.665	1.000	1.000	1.000	1.007
1.91	135.0	89.7	53.2	0.662	1.000	1.000	1.000	1.001
3.04	136.8	89.6	55.0	0.673	1.000	1.000	1.000	1.000
3.19	133.4	90.3	51.8	0.658	1.000	1.000	1.000	1.000
3.48	133.3	90.1	50.6	0.654	1.000	1.000	1.000	1.000